61:015

OAK RIDGE NATIONAL LABORATORY

OPERATED BY

UNION CARBIDE CORPORATION

NUCLEAR DIVISION

POST OFFICE BOX X

OAK RIDGE, TENNESSEE 37830

ORNL

CENTRAL FILES NUMBER

ORNL/CF-61-6-120

DATE: June 20, 1961

SUBJECT: MSRE Component Design Report
(MSR 61-67 of the MSRP series)

TO:

FROM: E. S. Bettis

INTRA-LABORATORY CORRESPONDENCE

Oak Ridge National Laboratory

June 20, 1961

MSR-61-67

To: Distribution

From: E. S. Bettis

Subject: MSRE Component Design Report

Distribution

S. E. Beall
M. Bender
E. S. Bettis
A. L. Boch
S. E. Bolt
W. L. Breazeale
R. B. Briggs
W. H. Ford
W. R. Grimes
A. G. Grindell
L. N. Howell
R. J. Kedl
H. G. MacPherson
W. B. McDonald
R. L. Moore
J. C. Moyers
C. W. Nestor
L. F. Parsly
H. R. Payne
R. E. Ramsey
D. Scott
I. Spiewak
A. Taboada
J. R. Tallackson
W. C. Ulrich
D. W. Vroom
J. H. Westsik

FOREWORD

These reports are submitted as justification of the design of components for the Molten-Salt Reactor Experiment. This reactor experiment is being run to establish and verify the operability, safety, and serviceability of a molten-salt reactor system which will be used as a basis for future power reactors.

The reports are written by the engineers having design responsibility for the various parts of the reactor system. An effort has been made in each case to present the calculations upon which the design is predicated in a concise and lucid manner. No attempt has been made to achieve literary perfection. Indeed the descriptive material has been kept to a minimum and an effort has been made to present the analytical work in rather complete detail.

It will be noticed that the analyses primarily concern design point calculations. Off design point studies are not included but will be used later to impose operational limitations on the system. The purpose of issuing the design reports in this form is to make available to interested technical persons the bases for the design of the components prior to actual initiation of fabrication of these items.

Every attempt has been made to devise a conservatively designed system. No component approaches high performance but the object has been to emphasize ruggedness and conservatism throughout. Consistency of design has also been attempted. One exception is encountered with the freeze flanges which, in the interest of conserving space, were made adequate for the calculated loading but not equal to the theoretical allowable load of the piping to which they are attached.

Questions about the design report can be directed to the individual authors of the reports or to the MSRE design section. Criticism of the results or methods employed in the analyses is invited. It is the intention of the MSRE Project to construct a reactor system which will conclusively demonstrate the adequacy of the fluoride reactor system as a potential power reactor concept.

Table of Contents

I. Mechanical Design of the MSRE Core No. 1

(H. R. Payne, W. H. Ford, J. C. Moyers, H. S. Weber)

II. Predicted Flow Distribution and Temperature Distribution in the MSRE Core

(R. J. Kedl)

III. Temperature Distributions in Various Regions of the Reactor Vessel for MSRE (MSR-61-62 dated 6-5-61)*

(W. L. Breazeale)

IV. Drain Tanks

(J. C. Moyers, H. R. Payne, J. H. Westsik)

V. Stress Analysis of Primary Heat Exchangers of the MSRE

(J. H. Westsik)

VI. Heat Transfer and ΔP Design of MSRE Primary Heat Exchanger (ORNL CF-61-4-1 dated 4-3-61)**

(J. H. Westsik)

VII. Radiator Design Calculations

(W. C. Ulrich)

VIII. MSRE Reactor Cell Overhead Biological Shield Requirements

(D. W. Vroom)

IX. Reactor Cell Analysis

(R. E. Ramsey)

* Limited distribution made previously. Copy is included herewith only to individuals not on earlier distribution list.

** Full distribution made previously.

I.

I. MECHANICAL DESIGN OF THE MSRE CORE NO. 1

MECHANICAL DESIGN OF THE MSRE CORE NO. 1

H. R. Payne
W. H. Ford - J. C. Moyers - H. S. Weber

INTRODUCTION

The MSRE core No. 1 was described and calculations to determine the reactor vessel shell and head thickness were reported in MSR-60-48. Since the publication of this letter a number of changes were made in the design of the reactor, and new data on the allowable design stress for INOR-8 were published. This core is redescribed here and the calculations are revised.

The reactor vessel shell and head thickness requirements are determined by the rules of the ASME Unfired Pressure Vessel Code Section VIII. The thermal stress in the shell and head and other critical parts of the reactor, and the mechanical stress or material thickness required to limit the stress to an acceptable value in these parts are determined. The control rods[1] are operated by an electric motor through a flexible cable and are air-cooled.

Appendix A is a list of the design drawings for the MSRE core No. 1.

DESCRIPTION OF THE CORE

The graphite moderator is composed of long (≈63-1/2-in.) blocks having a 2-in. x 2-in. "square cross section" with fuel passages having a 1.2-in. x 0.2-in. cross section cut in the four sides and fillets to eliminate the sharp corners in the passages. These blocks placed together give a fuel passage 1.2 in. x 0.4 in. (fillets in the corners). The graphite-to-fuel ratio is ~.225. The blocks rest on a graphite lattice and are contained in a 1/4-in.-thick INOR-8 shell (reactor core shell). The lower end of the core block are dowel shaped and extend through the graphite lattice. The holes in the lattice are sized to allow both angular and lateral movement between the core blocks and lattice. The height of the core blocks and lattice is 65-1/2 in. and the diameter of the bundle is 55-1/4 in. The 1/4-in.-thick shell containing the bundle is 55-1/2 in. ID.

The core blocks are restrained from excessive bowing by five molybdenum bands spaced over an 18-in.-long section, symmetrical about the midplane. A sixth band at the top will retain the blocks in a close-packed bundle should the others fail.

The core blocks on two diameters 90° apart extend 2 in. above the others except for five at the center. INOR-8 bars placed alongside these two rows, fastened to the shell at the outer end, keep the core blocks centered in the shell. An 8-in. square frame at the center provides access to five blocks at the center to be removed as samples. ~~The INOR-8 slab neutron shield which had been placed above the core block has been removed.~~

The reactor vessel is made of INOR-8. It is 58 in. ID x 9/16 in. thick and uses standard torispherical heads 60 in. OD x 1 in. thick. This container encloses the reactor core shell and provides an annular flow passage around the core shell for cooling it.

Fuel enters the reactor vessel through perforations in an 8-in. section of the reactor vessel wall at the top of the core shell, the inlet to the annular space between core shell and reactor vessel. The perforations are spaced to provide good flow distribution to the annulus (spacings checked experimentally), and the perforated plate is of sufficient thickness to provide the necessary strength to the vessel. The perforations are enclosed by the outer half of a torus having the DD ~~radii~~ diameter of an 8-in. diameter sched-40 pipe. The 5-in. sched-40 pipe of the primary system joins the semi-torus tangentially.

The fuel flows downward in a spiraling motion to the lower vessel head. The spiraling motion is stopped before it enters the graphite by radial vanes spaced around the outer periphery of the lower head. The length of the vanes is approximately 1/3 the radius of the vessel. The velocity and pressure drop in the several regions of the reactor are reported by R. J. Kedl.[2]

The core outlet is a 10-in. x 5-in. sched-40 reducing tee. The fuel leaving the reactor vessel enters the 10-in. run of the tee vertically and exits through the 5-in. branch horizontally. The 10-in. run (10-in. sched-40 pipe) terminates at a flange outside the thermal shield. A flanged plug fills the 10-in. run above the 5-in. branch.

A thin annulus between the plug and pipe in which fuel salt is frozen provides a seal on the reactor in addition to the ring joint of the flange.

Through the 10-in. plug are three thimbles and a 2-1/2-in. sched-40, plugged access with frozen fuel in the annulus similar to the 10-in. plug. The thimbles are 2-in.-OD x 0.065-wall tube welded into the 10-in. plug at the top, the capped ends extending to the lower end of the core blocks. These thimbles are penetrations for the three canned B_4C control rods discussed by E. S. Bettis[1] in MSR-61-53. ~~The poison tube described in MSR-61-48 has been eliminated in favor of three solid control rods.~~

The rods are normally air-cooled; and calculations are presented for the air required to limit the temperature of the flexible hose, on which the rod segments are threaded, to 1275°F.

Removal of the 10-in. plug provides access to the five full-length graphite core block samples. Four additional samples 7/8 OD x 33 in. long can be taken by removal of the 2-1/2-in. plug. These four samples occupy the place of the upper end of one of the 2-in. x 2-in. core blocks.

I. Mechanical Design of the Reactor Vessel Heads and Shell

~~Total~~ Weight of vessel
Weight of graphite
Weight of salt
Total

1. Design conditions:

Pressure	-	50 psig
Temperature	-	1300°F
S_a	-	2750 psi (allowable stress)

2. A_n = Area; n = 1, 2, ---
   - E = Joint efficiency; E = 1
   - L = Inside radius of dish
   - P = Design pressure
   - W = Axial load in the shell per inch of circumference
   - R = Inside radius of shell, head skirt, outlet nozzle
   - S_a = Allowable stress
   - S_t = Axial stress in vessel shell at the flow distributor
   - s = Arc length of region between holes in the flow distributor
   - t = Thickness of material required

3. Thickness of the shell:[3]

$$t = \frac{PR}{S_aE - 0.6P}$$

$$t = \frac{50 \times 29}{2750 - .6 \times 50} = 0.533 \text{ in.}$$

Use 9/16-in.-thick plate.

4. Thickness of heads:[3]

$$t = \frac{PL}{S_aE - .1P}$$

$$t = \frac{50 \times 54}{2750 - .1 \times 50} = .984 \text{ in.}$$

Use 1-1/8-in.-thick plate to insure a minimum thickness of 1 in. at the knuckle.

5. Reactor vessel outlet reinforcement (10-in. sched-40):

A. Thickness required in spherical region of the head:[3]

$$t = \frac{PL}{2S_a - .2P}$$

$$t = \frac{50 \times 54}{2 \times 2750 - .2 \times 50} = .492 \text{ in.}$$

B. Area of reinforcement required:[3]

$$A = 2Rt$$

$$A = 2 \times 5.02 \times .492 = 5.117 \text{ in.}^2$$

C. Nozzle thickness required:[3]

$$t = \frac{PR}{S_aE - 0.6P}$$

$$t = \frac{50 \times 5.02}{2750 \times 0.6 \times 50} = 0.092 \text{ in.}$$

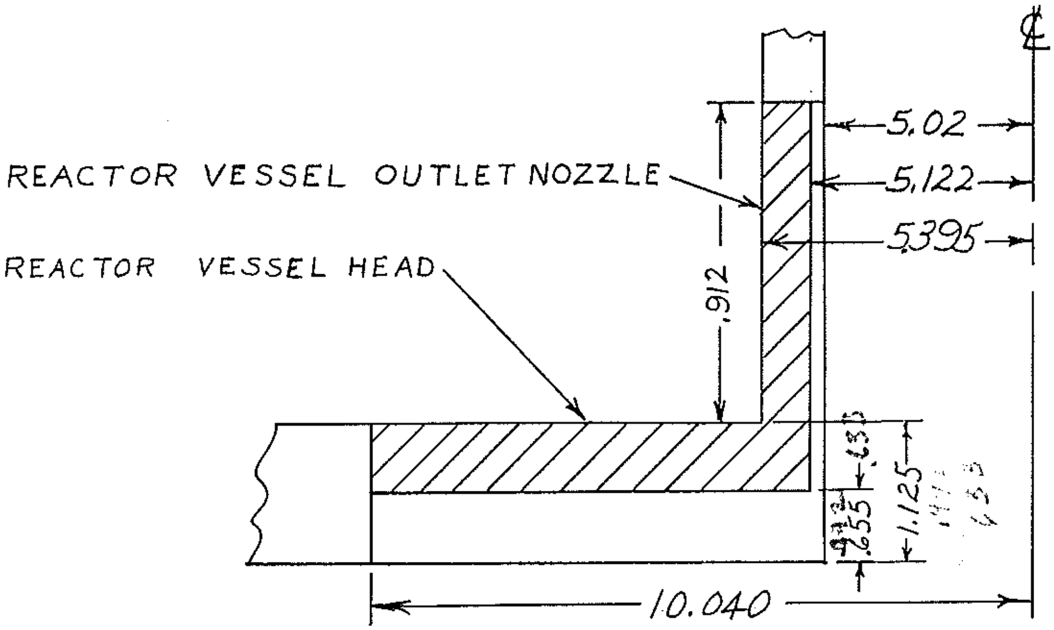


Fig. 1. Reactor Vessel Outlet Nozzle Reinforcement Area

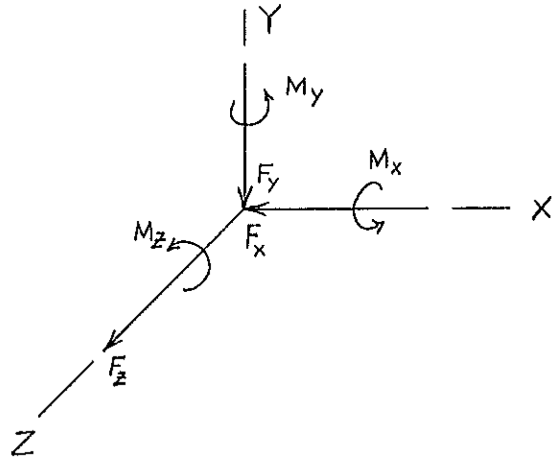
D. Reinforcement area available without adding additional metal (Fig. 1).[3]

$$A = 2(10.040 - 5.02) \times .653 + 2 \times .912(5.395 - 5.122)$$
$$= 1.306 \times 5.02 + 1.824 \times .273 = 7.053 \text{ in.}^2$$

6. Stress in the reactor vessel outlet nozzle due to piping reactions and internal heat generation:

Both the mechanical and thermal stresses are determined where the outlet tee joins the vessel head and two cases are considered. One, the reactor is isothermal (zero power); two, the reactor is at full power (10 Mw). The following analysis was made by H. S. Weber. Figure 2 depicts the reactions at the reactor outlet tee 7 in. above the joint between the outlet tee and the vessel head nozzle.

Reactor Isothermal - Case 1



F_x = -81 lb
F_y = -36 lb
F_z = 134 lb
M_x = 488 ft-lb
M_y = 288 ft-lb
M_z = 297 ft-lb

Fig. 2. Reactions at the Vessel Outlet--Isothermal Case

A. Bending moment with respect to horizontal ~~plant~~ plane at the joint:

a. $M_z' = F_x \ell + M_z$

b. $M_z' = 81 \times \frac{7}{12} + 297 = 344$ ft-lb

c. $M_x' = F_z \ell + M_x$

d. $M_x' = 134 \times \frac{7}{12} + 488 = 565$ ft-lb

e. The resultant moment is

$$M_R = \sqrt{(M_z')^2 + (M_x')^2}$$

f. $M_R = \sqrt{(344)^2 + (565)^2} = 661$ ft-lb

B. The axial bending stress due to M_R is

$$S_b = \frac{Mc}{I} = \frac{M_R}{\frac{\pi}{4R}(R^4 - r^4)}$$

$$S_b = \frac{661 \times 12 \times 4 \times 5.385}{\pi(5.385^4 - 5.01^4)} = 462 \text{ psi}$$

C. The axial stress due to F_y is:

$$S_{a1} = \frac{F_y}{\pi(R^2 - r^2)}$$

$$S_{a1} = \frac{-36}{\pi(5.385^2 - 5.01^2)} = -3 \text{ psi (compression)}$$

D. The stress due to the design pressure is:

a. tangential (circumferential)

$$S_t = \frac{PD}{2t}$$

$$S_t = \frac{50 \times 10.02}{2 \times .375} = 668 \text{ psi}$$

b. axial

$$S_{a2} = \frac{S_t}{2}$$

$$S_{a2} = \frac{668}{2} = 334 \text{ psi}$$

E. The shear stress is:

a. due to torsion

$$S_{s1} = \frac{16\, M_y\, D}{\pi(D^4 - d^4)}$$

$$S_{s1} = \frac{16 \times 288 \times 10.77}{\pi(10.77^4 - 5.01^4)} = 57 \text{ psi}$$

b. due to F_x and F_z

$$S_{s2} = \frac{\sqrt{(F_x)^2 + (F_z)^2}}{2t\ I} \times \frac{2}{3}(R^3 - r^3) =$$

$$\sqrt{(F_x)^2 + (F_z)^2}\ \frac{4}{3\pi t}\ \frac{(R^3 - r^3)}{(R^4 - r^4)}$$

$$S_{s2} = \sqrt{(81)^2 + (134)^2} \times \frac{1.33}{\pi x.375} \left(\frac{5.385^3 - 5.01^3}{5.385^4 - 5.01^4}\right)$$

$$= 26 \text{ psi}$$

F. Combining the stresses, we have

a. axial

$$\sigma_a = S_{a1} + S_{a2} + S_b$$

$$\sigma_a = -3 + 334 + 462 = 793 \text{ psi}$$

b. tangential

$$\sigma_t = S_t$$

$$\sigma_t = 668 \text{ psi}$$

c. shear

$$\sigma_s = S_{s1} + S_{s2}$$

$$\sigma_s = 57 + 26 = 83 \text{ psi}$$

G. The principal stresses are determined as follows

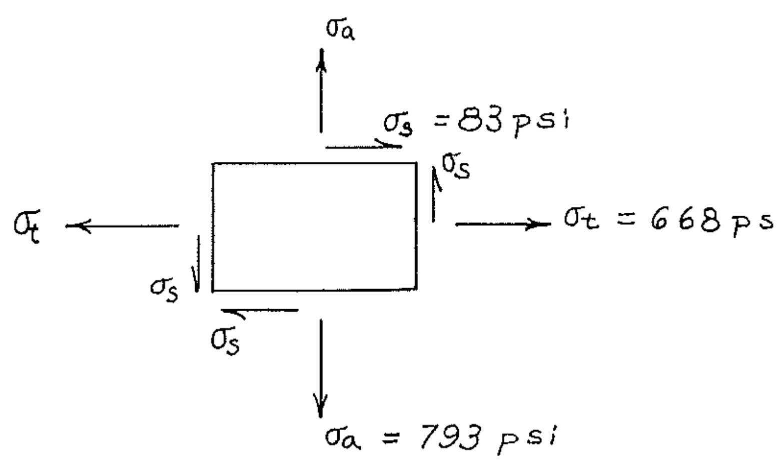


Fig. 3. Stress in the Outlet Nozzle at the Junction with the Outlet Tee

a. The maximum normal stress is

$$\sigma_n = \frac{\sigma_a + \sigma_t}{2} + \sqrt{\frac{(\sigma_t - \sigma_a)^2}{2} + \sigma_s^2}$$

$$\sigma_n = \frac{793 + 668}{2} + \sqrt{\frac{(793 - 668)^2}{2} + (83)^2}$$

$$\sigma_n = 731 + 104 = 835 \text{ psi}$$

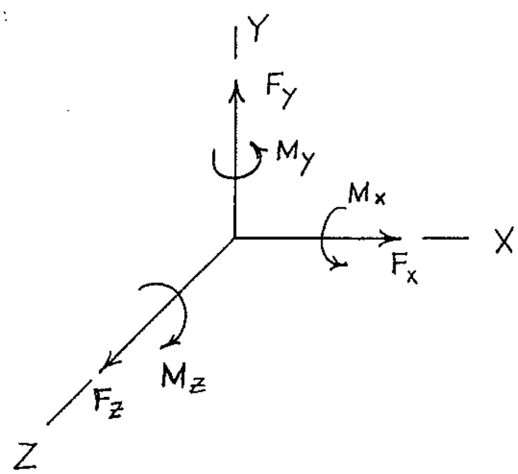
b. The maximum shear stress is

$$\sigma_s' = \sqrt{\frac{(\sigma_t - \sigma_a)^2}{2} + \sigma_s^2}$$

$$\sigma_s' = \sqrt{\frac{(793 - 668)^2}{2} + (83)^2} = 104 \text{ psi}$$

There are no thermal stresses in the isothermal condition.

Reactor at Full Power--Case 2



F_x = 26 lb

F_y = 51 lb

F_z = 81 lb

M_x = 69 ft-lb

M_y = 603 ft-lb

M_z = -170 ft-lb

Fig. 4. Reaction at the Vessel Outlet--Full Power Case

H. The moments and stresses determined as in the preceding case (reactor isothermal) give the following results.

a. M_R = 221 ft-lb

b. S_b = 154 psi

c. S_{a1} = 4 psi

d. S_t = 668 psi

e. S_{a2} = 334 psi

f. S_{s1} = 120 psi

g. S_{s2} = 15 psi

h. σ_a = 492 psi

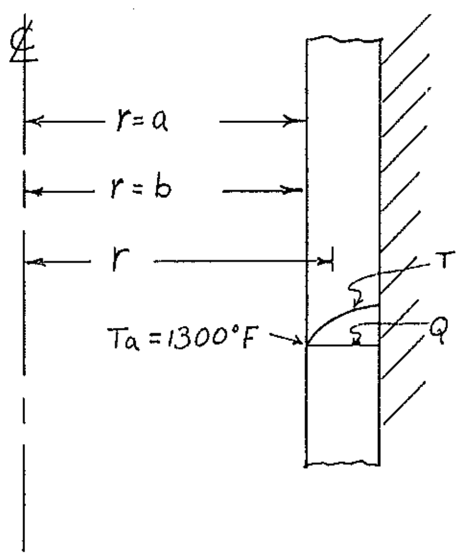
i. σ_t = 668 psi

j. σ_s = 135 psi

k. σ_n = 741 psi (maximum normal)

l. σ_s' = 161 psi (maximum shear)

The thermal stress is determined as follows, assuming the heat generated is transferred inward radially and the heat generation rate is uniform across the wall.



$r = a$ $T = 1300°F$

$r = b$ $\frac{dT}{dr} = 0$

Fig. 5. Reactor Vessel Outlet Nozzle Thermal Stress Model

Thermal stresses are determined for case 2 and combined with the mechanical stresses.

I. The temperature distribution through the wall is given by the following equation.[4]

$$T = 1300 - \frac{Q}{4K} + \frac{Qb^2}{2K} \ \ln \frac{r}{a}$$

J. The tangential stress is given by the following.[5]

$$\sigma_\theta = \frac{\alpha E}{(1-\nu)r^2} \left\{ \frac{r^2 + a^2}{b^2 - a^2} \int_a^b Trdr + \int_a^r Trdr - T^2 \right\}$$

K. a = 5.010-in. inside radius of nozzle

b = 5.385 in. outside radius of nozzle

E = 24.7×10^6

K = 13.05 Btu-ft/hr-ft^2-°F

Q = 54.13×10^3 Btu/hr-ft^3

r = radius at any point between a and b

T = temperature at r

α = 8×10^{-6} in./in.-°F

ν = .3

L. The tangential stress at r = b is

$$\sigma_{\theta b} = \frac{\alpha E}{(1-\nu)} \left\{ \frac{2}{b^2-a^2} \int_a^b \left[1300 - \frac{Q}{4K} + \frac{Qb^2}{2K} \ln \frac{r}{a} \right] rdr - \left(1300 - \frac{Q}{4K} + \frac{Qb^2}{2K} \ln \frac{b}{a} \right) \right\}$$

Integrating and substituting from the above values give

$$\sigma_{\theta b} = -423 \text{ psi}$$

M. At r = a, the tangential stress is

$$\sigma_{\theta b} = \frac{\alpha E}{(1-\nu)a^2} \left\{ \frac{2a^2}{b^2 - a^2} \int_a^b \left[1300 - \frac{Q}{4K} + \frac{Qb^2}{2K} \ln \frac{r}{a} \right] - \left(1300 - \frac{Q}{4K} \right) \right\}$$

Integrating and substituting from the above values gives

$$\sigma_{\theta a} = 395 \text{ psi}$$

N. The axial stresses determined in the same manner give the following values.

a. $\sigma_{za} = 395$ psi

b. $\sigma_{zb} = -423$ psi

O. The mechanical stresses are combined with the thermal stresses at r = a, to give the following maximum normal and shear stress.

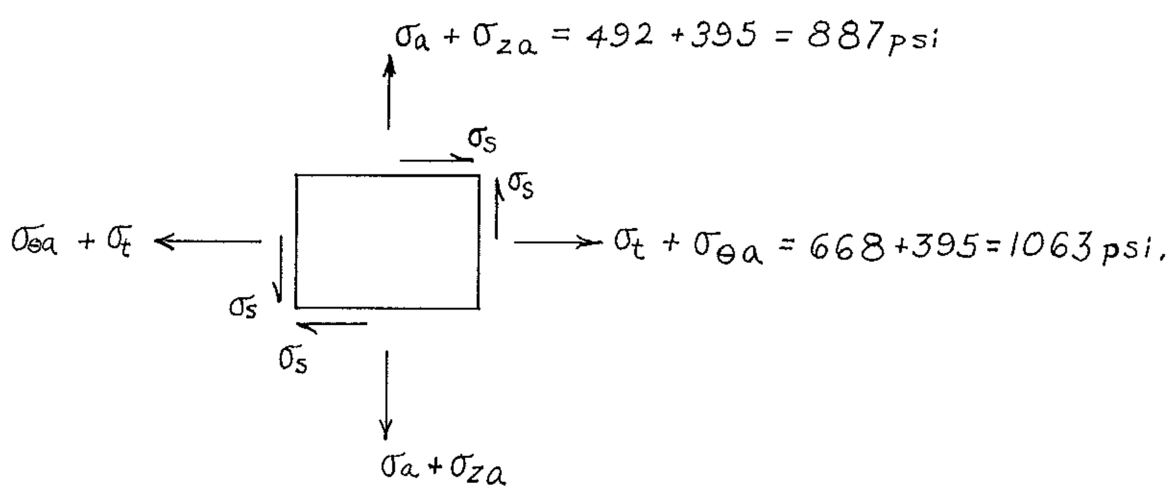


Fig. 6. Combined Thermal and Mechanical Stresses at r = a in the Reactor Vessel Outlet Nozzle

a. The maximum normal stress is

$$\sigma_n = \frac{\sigma_a + \sigma_{za} + \sigma_t + \sigma_{\theta a}}{2} + \sqrt{\frac{(\sigma_a + \sigma_{za} + \sigma_t + \sigma_{\theta a})^2}{2} + (\sigma_s)^2}$$

$$\sigma_n = \frac{887 + 1063}{2} + \sqrt{\frac{(887 + 1063)^2}{2} + (135)^2}$$

$$\sigma_n = 1136 \text{ psi}$$

b. The maximum shear is

$$\sigma_s' = \sqrt{\frac{(\sigma_a + \sigma_{za} + \sigma_t + \sigma_{\theta a})^2}{2} + (\sigma_s)^2}$$

$$\sigma_s' = \sqrt{\frac{(887 + 1063)^2}{2} + (135)^2}$$

$$\sigma_s' = 161 \text{ psi}$$

7. Flow distributor region of the shell

A. Assume uniform loading per inch of circumference (w) determined as follows. (See Fig. 7)

$$w = \frac{\pi R_1^2 P}{2\pi R} = \frac{RP}{2}$$

$$w = \frac{50 \times 29.5}{2} = 737.5 \text{ lb/in.}$$

B. The axial stress in the smallest areas is

$$S = \frac{w_s}{A_1}$$

$$S = \frac{737.5 \times (5\pi/180) \times 29.9}{29.5 \times (5\pi/180) - (.75/\sin 30^\circ)} = 1055 \text{ psi}$$

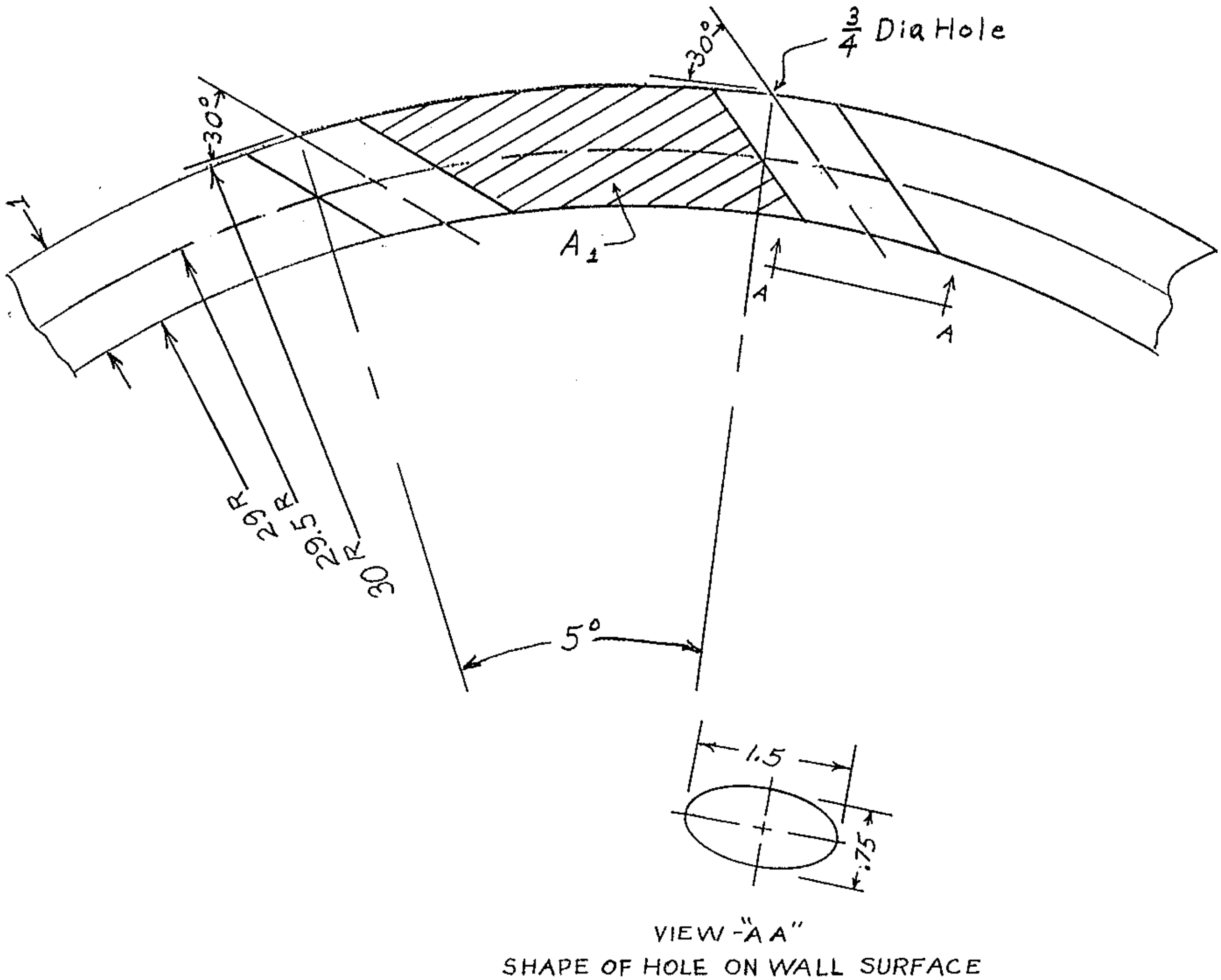
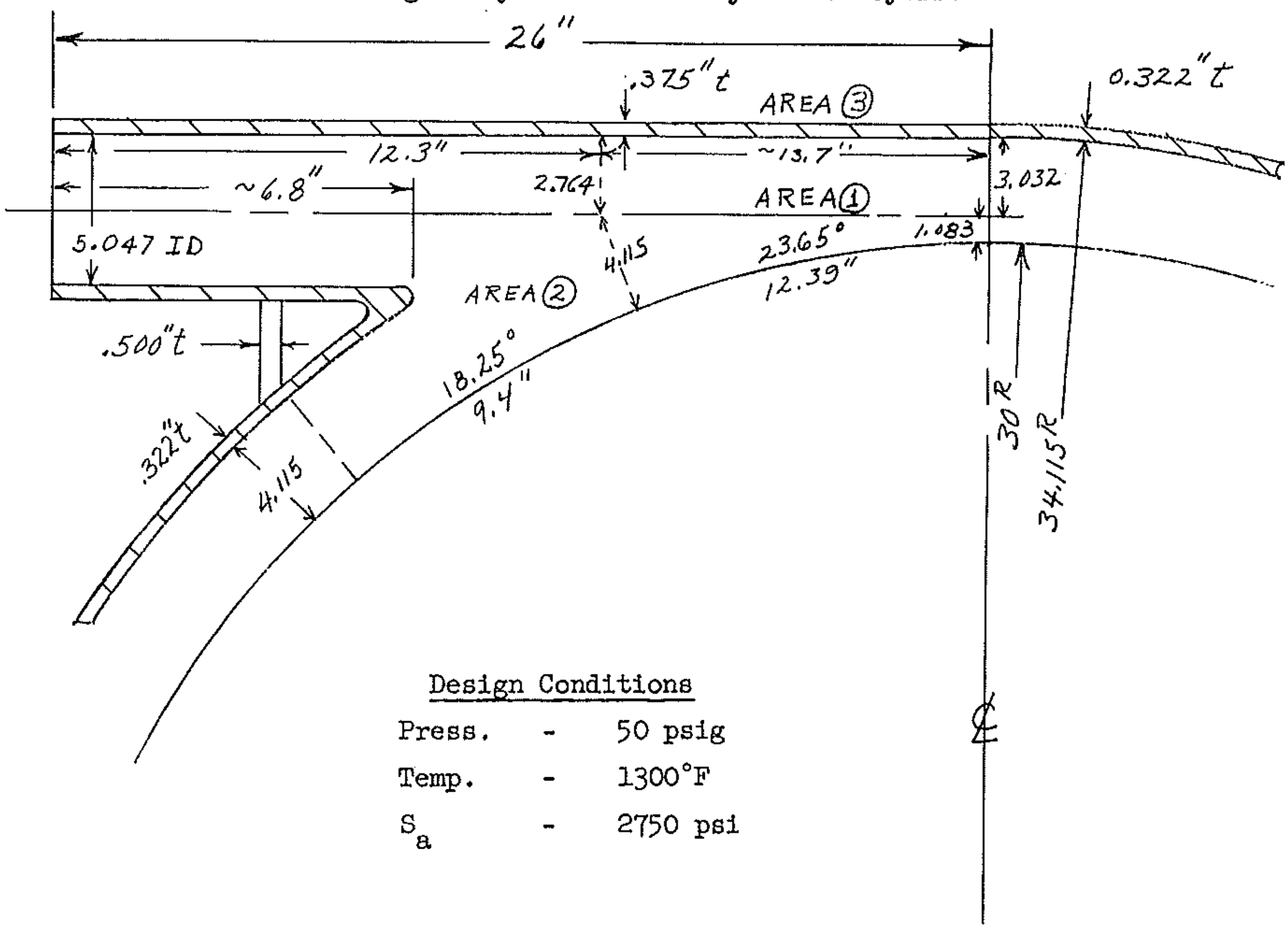


Fig. 7. Segment of a Cross Section of the Shell at the Flow Distributor Where Distribution Holes are Closest Spaced

8. Thermal and mechanical stress analysis of the reactor inlet nozzle.

The following analysis was made by J. C. Moyers.



Design Conditions

Press.	-	50 psig
Temp.	-	1300°F
S_a	-	2750 psi

Fig. 8. Horizontal Cross Section of the Reactor Inlet Nozzle

A. Analysis of Pressure Stresses - (Use "Design of Piping Systems," W. M. Kellog Co.)

$$\text{Area } ① \approx 13.7 \times 2.764 + \frac{13.7 + 12.39}{2} \times \frac{4.115 + 1.083}{2}$$

$$\approx 71.8 \text{ in.}^2$$

$$\text{Area } ② \approx \pi(34.115^2 - 30^2)\left(\frac{18.25}{360}\right) + \frac{1}{2} \times 5.5 \times 2.657 + 6.8 \times 2.590$$

$$\approx 67.1 \text{ in.}^2$$

Area (3) $= 13.7 \times 0.375 = 5.14$ in.^{2}

Area (4) $\cong (6.8)(.375) + 4.75(.322)$

$\cong 4.08$ in.^{2}

$$S_3 = \frac{P\,(A_{(1)} + 1/2\,A_{(3)})}{A_{(3)}} = \frac{50\,(71.8 + 2.57)}{5.14}$$

$= 723$ psi (tangential tension)

$$S_4 = \frac{P\,(A_{(2)} + 1/2\,A_{(4)})}{A_{(4)}} = \frac{50\,(67.1 + 2.04)}{4.08}$$

$= 847$ psi (tangential tension)

Longitudinal pressure stress in the nozzle:

$$S = \frac{PD}{4T} = \frac{50 \times 5.047}{4 \times .375} = 168.3 \text{ psi (tension)}$$

<u>Piping Reactions on Nozzle</u> (furnished by Ramsey, 5-19-61)

<u>Isothermal Condition</u>

$F_x = +\,81$ lb $M_x = +702$ ft-lb

$F_y = +\,36$ lb $M_y = +613$ ft-lb (bending)

$F_z = -134$ lb $M_z = +513$ ft-lb

Combined M_x and M_z (torsion) = 870 ft-lb (torsion)

<u>Operating Temperature</u>

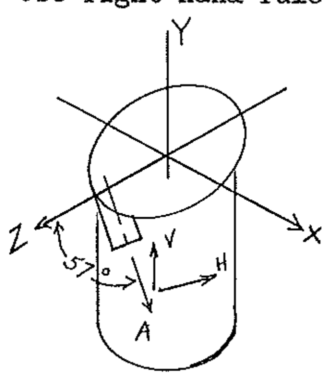
$F_x = -26$ lb $M_x = +460$ ft-lb

$F_y = -51$ lb $M_y = +547$ ft-lb (bending)

$F_z = -87$ lb $M_z = -254$ ft-lb

Combined M_x and M_z = 525 ft-lb (torsion)

Use right-hand rule for sign of moments.



<u>Axis Orientation</u>

<u>Transposing to axial & transverse axes</u>

$F_A = F_x \cos 33° + F_z \cos 57°$

$F_T = \sqrt{(F_y)^2 + (F_x \sin 33° - F_z \sin 57°)^2}$

It may be desirable to have transverse force in its horizontal and vertical components - $F_H = F_x \sin 33° - F_z \sin 57°$

$F_V = F_y$

For Isothermal Case:	For Operating Temperature Case:
$F_A = +81 \cos 33° + (-134)\cos 57°$	$F_A = (-26)\cos 33° + (-87)\cos 57°$
$= 67.9 - 73$	$= -21.8 - 47.4$
$= -5.1$ lb (compressive)	$= -69.2$ lb (compressive)
$F_H = 81 \sin 33° - (-134)\sin 57°$	$F_H = (-26)\sin 33° - (-87)\sin 57°$
$= 44.1 + 112.5$	$= -14.2 + 73$
$= +156.6$ lb	$= +58.8$ lb
$F_V = +36$ lb	$F_V = -51$ lb
$F_T = \sqrt{(36)^2 + (156.6)^2} = 160.7$ lb	$F_T = \sqrt{(58.8)^2 + (-51)^2} = 77.9$ lb

Stresses Due to Piping Reactions on Nozzle

B. Stress due to axial thrust of piping:

Axial load will be distributed elastically along the welded juncture of the cone and the torus, so the stress from this load should become insignificant shortly after the junction starts. The minimum metal area where the entire load appears is at the nozzle end. The stress at this point is

$$S_{max_{o.t.}} = P/A = \frac{-69.2 \text{ lb}}{6.34 \text{ in.}^2} = -\underline{\underline{10.9 \text{ psi}}} \text{ (longitudinal compressive)}$$

$$S_{max_{iso}} = -.8 \text{ psi}$$

C. Stress due to horizontal transverse reaction and Y moment

Check of stress at strut -

Assume zero rigidity of nozzle at juncture with torus (i.e.- strut carries all the load). This is ultra conservative.

$$\text{Compressive stress in strut} = \frac{156.6}{0.5 \text{x} 5} \times \frac{12.3}{9.5} = 81.1 \text{ psi (iso.)}$$

30.5 psi (o.t.)

$$\text{Shear stress in nozzle at strut} = \frac{156.6}{11.8 \text{x}.375} \times \frac{12.3}{9.5} = 45.8 \text{ psi (iso.)}$$

17.2 psi (o.t.)

Bending stress due to reaction and Y moment

Ignore strut and take stress at beginning of juncture (L = 6.8 in.)

For isothermal case, bending moment $= 613 + \frac{6.8}{12} \times 156.6$

$= 702$ ft lb

For operating temperature case, moment $= 547 + \frac{6.8}{12} \times 58.8$

$= 580$ ft lb

Bending stress $= \frac{Mc}{I}$

$I = \frac{\pi}{4}(3.032^4 - 2.657^4) = 27.6 \text{ in.}^4$

$c = 3.032$ in.

Bending stress $= \frac{702 \times 12 \times 3.032}{27.6} = 927$ psi (iso) (tension on outside) (compression on inside)

$= 766$ psi (o.t.)

Check of stress at pipe-nozzle juncture (smaller moment but smaller pipe)

$I = \frac{\pi}{4}(2.899^4 - 2.524^4) = 23.5 \text{ in.}^4$

$c = 2.899$

Bending stress $= \frac{613 \times 12 \times 2.899}{23.5} = 948$ psi (iso) (tension outboard compression inboard)

$= 846$ psi (o.t.) (tension outboard compression inboard)

D. Bending stress due to Y reaction:

For isothermal case:

$$S = \frac{36 \times 6.8 \times 3.032}{27.6} = 26.9 \text{ psi}$$ (compression top, tension bottom)

For operating temperature case:

$$S = \frac{51}{36} \times 26.9 = 38.1 \text{ psi}$$ (tension top, compression bottom)

E. Shear stress due to torsion from piping:

$$S_s = \frac{T}{Z_o}$$

$$Z_o = \frac{\pi(D_o^4 - D_i^4)}{16 D_o}$$

$$Z_o(\text{end}) = \frac{\pi(5.797^4 - 5.047^4)}{16(5.797)} = 16.25 \text{ in.}^3$$

$$Z_o(\text{juncture}) = \frac{\pi(6.064^4 - 5.314^4)}{16(6.064)} = 18.08 \text{ in.}^3$$

For the isothermal case:

$$S_s(\text{end}) = \frac{870 \times 12}{16.25} = 642 \text{ psi}$$

$$S_s(\text{juncture}) = \frac{870 \times 12}{18.08} = 578 \text{ psi}$$

For operating temperature case:

$$S_s(\text{end}) = \frac{525 \times 12}{16.25} = 388 \text{ psi}$$

$$S_s(\text{juncture}) = \frac{525 \times 12}{18.08} = 348 \text{ psi}$$

F. Shear stress due to transverse pipe reactions:

Nozzle shear area at end $= \pi(2.899^2 - 2.524^2) = 6.34 \text{ in.}^2$

Nozzle shear area at junction $= \pi(3.139^2 - 2.764^2) = 6.91 \text{ in.}^2$

For the isothermal case:

$$S_s(\text{end}) = \frac{160.7}{6.34} = 25.4 \text{ psi}$$

$$S_s(\text{juncture}) = \frac{160.7}{6.91} = 23.3 \text{ psi}$$

For the operating temperature case:

$$S_s(\text{end}) = \frac{77.9}{6.34} = 12.3 \text{ psi}$$

$$S_s(\text{juncture}) = \frac{77.9}{6.91} = 11.3 \text{ psi}$$

G. Thermal stresses due to γ heating:

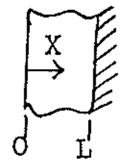
γ heating rates (supplied by Nestor, 6-1-61):

Outside vessel at midplane = 0.0461 watts/cc

Outside vessel at top plane = 0.0131 watts/cc

To be conservative, since the nozzle is at an intermediate location, use the midplane value (0.0461 w/cc = 2.33 Btu/in.^{3}).

$$\frac{\partial^2 T}{\partial X^2} = -\frac{Q}{K}\ ; \qquad \frac{\partial T}{\partial X} = -\frac{QX}{K} + C_1\ ; \qquad T = -\frac{QX^2}{2K} + C_1 X + C_2$$



This assumes slab geometry and uniform internal heat generation. Assuming the outside of the pipe perfectly insulated, the boundary conditions are:

$$\frac{\partial T}{\partial X} = 0 \text{ @ } X = L \quad \text{and} \quad T = T_o = \text{datum} = 0 \text{ @ } X = 0$$

$$C_1 = \frac{QL}{K}\ ; \quad C_2 = 0$$

Then $\quad T = \frac{Q}{K}\left(LX - \frac{X^2}{2}\right)$

Tangential thermal stress, σ_θ

$$= \frac{\alpha E}{1-\nu}\ \frac{1}{r^2}\left(\frac{r^2 + a^2}{b^2 - a^2}\int_a^b Tr\,dr + \int_a^r Tr\,dr - Tr^2\right)$$

Substituting the above expression for T and integrating,

$$\sigma_\theta = \frac{\alpha E}{1-\nu}\left(\frac{Q}{Kr^2}\right)\left(\frac{r^2+a^2}{b^2-a^2}\right)\left(\frac{Lb^3}{3} - \frac{Lab^2}{2} - \frac{b^4}{8} + \frac{ab^3}{3} - \frac{a^2b^2}{4} + \frac{La^3}{3}\right.$$

$$\left. + \frac{a^4}{12} - \frac{2Lr^3}{3} + \frac{Lar^2}{2} + \frac{3r^4}{8} - \frac{2ar^3}{3} + \frac{a^2r^2}{4}\right)$$

The maximum stress will occur at r = a since that is the cooler surface and the temperature gradient is steeper there.

$$\sigma_{\theta a} = \frac{\alpha EQ}{(1-\nu)K}\left(\frac{2}{b^2-a^2}\right)\left(\frac{Lb^3}{3} - \frac{Lab^2}{2} - \frac{b^4}{8} + \frac{ab^3}{3} - \frac{a^2b^2}{4} + \frac{La^3}{6} + \frac{a^4}{24}\right)$$

Assuming pipe mean temperature is 1300°F,

$\alpha = 7.98 \times 10^{-6}$ °F^{-1} | $K = 1.09$ Btu/in.-hr-°F

$E = 24.7 \times 10^{6}$ psi | $L = 0.375$ in.

$\nu = 0.3$ | $a = 2.524$ in.

$Q = 2.33$ Btu/in.^{3}-hr | $b = 2.899$ in.

Solving the above equation,

$\sigma_{\theta a} = 17$ psi

The axial stress, σ_z

$$= \frac{\alpha E}{1-\nu}\left(\frac{2}{b^2 - a^2}\int_a^b Trdr - T\right)$$

Substituting for T and integrating,

$$\sigma_z = \frac{\alpha E}{1-\nu}\left(\frac{2}{b^2-a^2}\right)\left(\frac{Q}{K}\right)\left(\frac{Lb^3}{3} - \frac{Lab^2}{2} - \frac{b^4}{8} + \frac{ab^3}{3} - \frac{a^2b^2}{4} + \frac{La^3}{6} + \frac{a^4}{24}\right.$$
$$\left. - Lr + La + \frac{r^2}{2} - ra + \frac{a^2}{2}\right)$$

At $r = a$, where the stress is maximum, $\sigma_z = \sigma_\theta = 17$ psi

H. Summary of stresses:

Isothermal case:

Source	Axial	Tangential	Shear
Pressure	+168.3	+847	-
Axial thrust	- 0.8	-	-
Hor. reaction + Y moment	±948	-	45.8
Bending from Y reaction	± 26.9	-	-
Piping torsion	-	-	642
Thermal	-	-	-
	+1147.4	+847	687.8

Operating temperature case:

Source	Axial	Tangential	Shear
Pressure	+168.3	+847	-
Axial thrust	- 10.9	-	-
Hor. reaction + Y moment	±846	-	17.2
Bending from Y reaction	± 38.1	-	-
Pipe torsion	-	-	388
Thermal	+ 17		
	+1058.5	+864	405.2

Combining these stresses:

For the isothermal case:

$$\text{Max normal stress} = \frac{1147 + 847}{2} + \sqrt{\left(\frac{1147 - 847}{2}\right)^2 + 688^2}$$

$$= 997 + \sqrt{475,594}$$

$$= 1687 \text{ psi}$$

Max shear = 690 psi

For the operating temperature case:

$$\text{Max normal stress} = \frac{1059 + 864}{2} + \sqrt{\left(\frac{1059 - 864}{2}\right)^2 + 405^2}$$

$$= 962 + \sqrt{173,531}$$

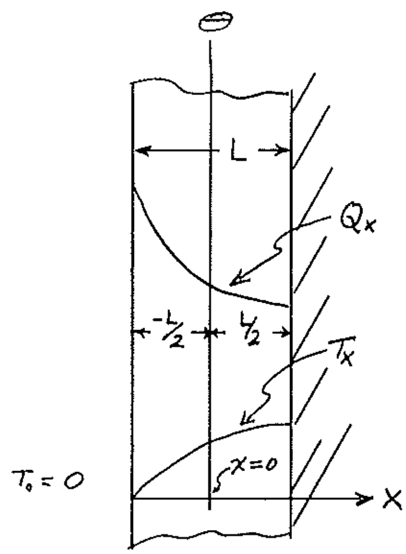
$$= 1379 \text{ psi}$$

Max shear = 417 psi

II. Thermal Stress in the Reactor Vessel Heads

The heads are treated as slabs with a variable internal heat generation rate of the following nature: $Q_x = Q_o e^{-\mu x}$. Two cases are considered. One, the head is perfectly insulated on the outside. Two, the inside and outside surfaces are at the same temperature.

Case 1 - heads perfectly insulated outside:



$x = -L/2$ $\quad$ $x = L/2$

$T = 0$ $\quad$ $\frac{\partial T}{\partial x} = 0$

Fig. 9. Reactor Vessel Head Thermal Stress Model
Case 1

A. $Q_x = Q_o e^{-\mu x}$

B. $\frac{\partial^2 T}{\partial x^2} = -\frac{Q_o}{K} e^{-\mu x}$ (ref. 4)

C. $T = T_x - T_o = \frac{Q_o}{K\mu}\left[\frac{e^{\mu L/2}}{\mu} - e^{-\mu L/2} x - \frac{Le^{-\mu L/2}}{2} - \frac{e^{-\mu x}}{\mu}\right]$

D. $\sigma_\theta = -\frac{\alpha TE}{(1-\nu)} + \frac{1}{2(L/2)(1-\nu)} \int_{-L/2}^{L/2} \alpha TE \, dx + \frac{3x}{2(L/2)^3(1-\nu)} \int_{-L/2}^{L/2} \alpha TE \, x \, dx$ (ref. 5)

E. $$\sigma_\theta = \frac{\alpha E Q_o}{(1-\nu)K\mu^2}\left[e^{-\mu x} + \frac{e^{-\mu L/2}}{\mu L} - \frac{e^{\mu L/2}}{\mu L} + \frac{6xe^{-\mu L/2}}{\mu L^2} + \frac{6xe^{\mu L/2}}{\mu L^2} + \frac{12\ xe^{-\mu L/2}}{\mu^2 L^3} - \frac{12\ xe^{\mu L/2}}{\mu^2 L^3}\right]$$

F. $Q_o = 37.46$ Btu/in.^{3}-hr

$L = 1.125$ in.

$\mu = .769$ in.$^{-1}$

$L^2 = 1.2656$ in.^{2}

$L^3 = 1.4238$ in.^{3}

$\mu^2 = .5914$ in.$^{-2}$

$\mu L = .8651$

$\mu^2 L^2 = .7485$

$\alpha = 8 \times 10^{-6} \frac{\text{in.}}{\text{in.-°F}}$

$E = 24.7 \times 10^6$

$K = 12.68 \frac{\text{Btu-ft}}{\text{hr-ft}^2\text{-°F}}$

$\nu = .3$

$e^{\mu L/2} = 1.541$

$e^{-\mu L/2} = .6489$

G. At $x = -L/2$ substituting from the above values

$$\sigma_\theta = \frac{8 \times 10^{-6} \times 24.8 \times 10^6 \times 37.46 \times 12}{.7 \times 12.5 \times (.769)^2}\left[e^{\mu L/2} - \frac{2e^{-\mu L/2}}{\mu L} - \frac{4e^{\mu L/2}}{\mu L} - \frac{6e^{-\mu L/2}}{\mu^2 L^2} + \frac{6e^{\mu L/2}}{\mu^2 L^2}\right]$$

$$= 14{,}139 \times \left[1.541 - \frac{2 \times .6489}{.8651} - \frac{4 \times 1.541}{.8651} - \frac{6 \times .6489}{.7485} + \frac{6 \times 1.541}{.7485}\right]$$

$$= 14{,}139 \times .067 = 946 \text{ psi}$$

H. At $x = 0$

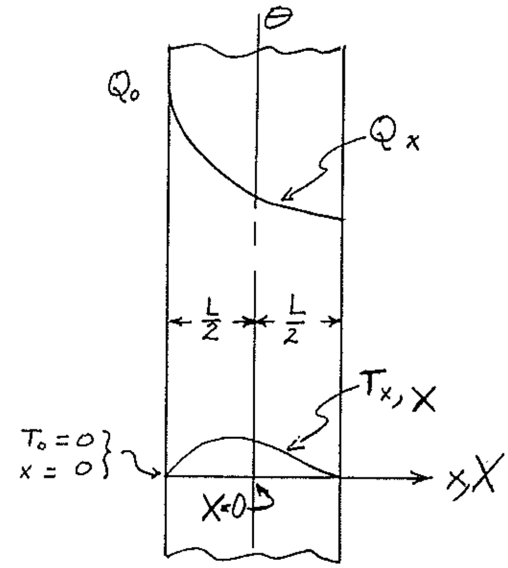
$$\sigma_\theta = 14{,}139\left[1 + .7501 - 1.7812\right] = -440 \text{ psi}$$

I. At $x = L/2$

$$\sigma_\theta = 14{,}139 \left[.6489 + \frac{4 \times .6489}{.8651} + \frac{2 \times 1.541}{.8651} + \frac{6 \times .6489}{.7485} - \frac{6 \times 1.541}{.7485} \right]$$

$$= 14{,}139 \times .0601 = 849 \text{ psi}$$

Case 2 - inside and outside surface at the same temperature



$x = 0 \quad x = L$

$T = 0 \quad T = 0$

Fig. 10. Reactor Vessel Head Thermal Stress Model
Case 2

J. $Q_x = Q_o e^{-\mu x}$

K. $\dfrac{\partial^2 T}{\partial x^2} = -\dfrac{Q_o}{K} e^{-\mu x}$ (ref. 4)

L. $T = (T_X - T_o) = \dfrac{-Q_o}{K\mu^2} \left[e^{-\mu(X+L/2)} - \dfrac{e^{-\mu L}}{L} X - \dfrac{e^{-\mu L}}{2} + \dfrac{X}{L} - \dfrac{1}{2} \right]$

M. $\sigma_\theta = \dfrac{-\alpha TE}{(1-\nu)} + \dfrac{1}{2L/2(1-\nu)} \displaystyle\int_{-L/2}^{L/2} \alpha ET \, dX + \dfrac{3X}{2(L/2)^3(1-\nu)} \int_{-L/2}^{L/2} \alpha TEXdX$ (ref. 5)

N. $$\sigma_\theta = \frac{Q_o \alpha E}{K\mu^2(1-\nu)}\left[e^{-\mu(X+L/2)} + e^{-\mu L}\left(\frac{1}{\mu L} + \frac{6X}{\mu L^2} + \frac{12X}{\mu^2 L^3}\right) - \frac{1}{\mu L} + \frac{6X}{\mu L^2} - \frac{12X}{\mu^2 L^3}\right]$$

O. At $X = -L/2$ substiting from the values section II-1.F

$$\sigma_\theta = 14{,}139\left[1 + e^{-\mu L} \quad \frac{-2}{\mu L} - \frac{6}{\mu^2 L^2} \quad - \frac{4}{\mu L} + \frac{6}{\mu^2 L^2}\right]$$

$$= 14{,}139 \times .0443 = 626 \text{ psi}$$

P. At $X = 0$

$$\sigma_\theta = 14{,}139\left[e^{-\mu L/2} + \frac{e^{-\mu L}}{\mu L} - \frac{1}{\mu L}\right]$$

$$= 14{,}139 \times (-.0203) = -287 \text{ psi}$$

Q. At $X = L/2$

$$\sigma_\theta = 14{,}139\left[e^{-\mu L} + e^{-\mu L}\left(\frac{4}{\mu L} + \frac{6}{\mu^2 L^2}\right) + \frac{2}{\mu L} - \frac{6}{\mu^2 L^2}\right]$$

$$= 14{,}139 \times .0482 = 681 \text{ psi}$$

III. Thermal Stress in the Reactor Vessel Shell

The pressure vessel shell is treated as a long cylinder having a variable internal heat generation rate of the following nature: $Q_r = Q_o e^{-\mu r}$. It is further assumed that all heat generated within the shell flows inward (shell perfectly insulated). The tangential and the axial stress is determined from the following equations.[7]

1. Tangential stress

$$\sigma_\theta = -\frac{Q_o \alpha E}{(1-\nu)K\mu^2 r^2}\left\{\frac{r^2+a^2}{b^2-a^2}\left[\frac{a^2-b^2}{2}+\frac{\mu a+1}{\mu^2}-e^{-\mu(b-a)}\left(\frac{\mu b+1}{\mu^2}-\frac{\mu b^3}{3}\right.\right.\right.$$

$$\left.\left.+\frac{\mu a b^2}{2}-\frac{\mu a^3}{6}\right)\right]-e^{-\mu(r-a)}\left[\frac{\mu r+1}{\mu^2}+r^2\right]-e^{-\mu(b-a)}\left[-\frac{\mu a r^2}{2}\right.$$

$$\left.\left.+\frac{2\mu r^3}{3}-\frac{\mu a^3}{6}\right]+\frac{a^2+r^2}{2}+\frac{\mu a+1}{\mu^2}\right\}$$

2. Axial stress

$$\sigma_z = -\frac{Q_o \alpha E}{(1-\nu)K\mu^2}\left\{\frac{2}{b^2-a^2}\left[\frac{a^2-b^2}{2}+\frac{\mu a+1}{\mu^2}-e^{-\mu(b-a)}\left(\frac{\mu b+1}{\mu^2}-\frac{\mu b^3}{3}+\frac{\mu a b^2}{2}\right.\right.\right.$$

$$\left.\left.\left.-\frac{\mu a^3}{6}\right)\right]-\left[e^{-\mu(r-a)}+\mu(r-a)e^{-\mu(b-a)}-1\right]\right\}$$

3. The stress is determined as follows.

A. $a = 29$ in. inside radius

$b = 29.562$ outside radius

r = Radius to any point in shell wall

$E = 24.7 \times 10^6$ lb/in.^{2}

$K = 12.68$ Btu-ft/hr-ft^2-°F

$Q_o = 3.92$ Btu/in.^{3}-hr

$\alpha = 8 \times 10^{-6}$ in./in.-°F

$\mu = .769$ in.$^{-1}$

$\nu = .3$

B. At $r = a$, substituting from above, the tangential stress is

$$\sigma_\theta = - \frac{3.92 \times 8 \times 24.7 \times 12}{.7 \times 12.68 \times .769^2 \times 29^2} \left\{ \frac{2 \times 29^2}{29.562^2 - 29^2} \left[\frac{29^2 - 29.562^2}{2} \right.\right.$$

$$+ \frac{.769 \times 29 + 1}{.769^2} - e^{-.769 \times .562} \left(\frac{.769 \times 29.5625 + 1}{.769^2} - \frac{.769 \times (29.562)^3}{3} \right.$$

$$\left.\left. + \frac{.769 \times 29 \times 29.5625^2}{2} - \frac{.769 \times 29^3}{6} \right) \right]$$

$$\sigma_\theta = -2.16 \left\{ 51.1063 \left[- 16.4559 + 39.3997 - .649 \, (40.1305 - 6{,}622.2642 \right.\right.$$

$$\left. + 9{,}744.5535 - 3{,}125.8568) \right]$$

$$\sigma_\theta = - 2.16 \times 51.1063 \times (-.7855) = 87 \text{ psi}$$

C. At $r = b$ the tangential stress is

$$\sigma_\theta = - 2.16 \left\{ 52.1063 \left[-16.4559 + 39.3997 - .649 \, (40.1305 - 6{,}622.2642 \right.\right.$$

$$\left. + 9{,}744{,}5535 - 3{,}125.8568) \right] - .649 \left[39.4396 + 873.9118 \right.$$

$$\left. - 9{,}744.5535 + 13{,}245.3895 - 3125.8568 \right] + 857.4559$$

$$\left. + 39.3997 \right\}$$

$$= - 2.16 \left\{ 52.1063 \times (-.7855) - 836.1265 + 896.8556 \right\}$$

$$= - 41 \text{ psi}$$

D. At $r = a$, substituting the values from section III-3.A into formula of section III-2, the axial stress is

$$\sigma_z = -\ 1821.437 \left\{ \frac{2}{32.9118} \left[-\ 16.4559 + 39.3997 - .649\ (40.1305 - 6{,}622.2642 + 9{,}744.5535 - 3{,}125.8568) \right] \right.$$

$$\sigma_z = -\ 1821.437 \times .0608 \times (-.7855) = 87 \text{ psi}$$

E. At $r = b$ the axial stress is

$$\sigma_z = -\ 1821.437 \left\{ .0608 \times (-.7855) - .649 - .280 + 1 \right\}$$

$$= -\ 43 \text{ psi}$$

IV. Design of the Moderator (Core) Supporting Structure

1. Core support-grid mechanical design

The height required in the grid bars for those bars carrying the greatest load is determined here. This same method was used to determine the height of the other bars.

area of the grids that carry the load = 316.5 in²
assumed load = 36.65 #/in²

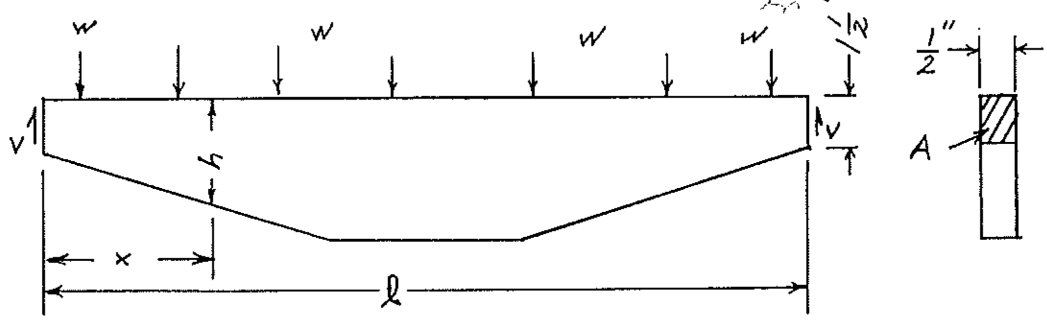


Fig. 11. Typical Core Support-Grid Bar

A. $A = .75\ \text{in.}^2$ cross section area at the ends

$h = \sqrt{\frac{12M}{S}}$, height of beam required at x

$\ell = 55$ in., length of beam

M = bending moment at x

S = 2750 psi, allowable bending stress at x

w = 14.2 lb/in., distributed load

x = distance to any cross section along the length

B. $M = \frac{1}{2}\, w\ell\left(x - \frac{x^2}{\ell}\right) = \frac{w}{2}\,(x\ell - x^2)$ (ref. 8)

C. $h = \sqrt{\frac{12M}{S}} = \sqrt{\frac{6w(\ell x - x^2)}{S}}$

x (in.)	h (in.)
0	0
3	2.21
6	3.03
9	3.6
12	3.99
15	4.32
18	4.55
21	4.71
24	4.82
27	4.85
27.5	4.85

$I = \frac{1}{12} bh^3$

Roark page 13

$h = \sqrt{\frac{6M}{bS}}$

$h = \sqrt{\frac{12M}{S}}$

$h = \sqrt{\frac{6w(\ell x - x^2)}{S}}$

D. The greatest shear stress occurs at the ends and is determined in the following manner.

??

$S_s = \frac{w\ell}{2A}$

$S_s = \frac{14.2 \times 55}{2 \times .5 \times 1.5} = 520$ psi

$$S = \frac{Mc}{I} = \frac{M\frac{h}{2}}{\frac{bh^3}{12}} = \frac{6M}{bh^2}$$

$$M = \frac{w}{2}(xl - l^2)$$

at $x = l/2$

$$M = \frac{w}{2}\left(\frac{l^2}{2} - \frac{l^2}{4}\right) = \frac{wl^2}{8}$$

$$M = \frac{(3.665 \times 10)(5.5 \times 10)^2}{8} = \frac{1.10866 \times 10^5}{8}$$

$$M = 1.385833 \times 10^4 \text{ in-\#}$$

$$S = \frac{6M}{bh^2} = \frac{6(1.385833 \times 10^4)}{(0.5)(5.125)^2} = \frac{8.31498 \times 10^4}{13.13282}$$

$$S = 6.33145 \times 10^3$$

$$S = 6331 \text{ psi}$$

$S = \frac{Mc}{I} = \frac{\quad}{h^2}$

$M = \frac{w}{2}(x l - x^2)$

at $\frac{l}{2}$: $M = 1.38583 \times 10^4$ $S = 6$[illegible]

The width of the bar where the bending stress is the greatest is 5.5 in. less .375 in. for core block anchoring rod. This gives a bending stress of 2460 psi. Due to the shape of the bar and the low shear stress the combined stress will not exceed an acceptable value.

Spacer blocks between the grid bars maintain the spacing between the bars and provide lateral rigidity to the structure.

2. Thermal stress in the core support-grid bars

A section $x = \ell/2$ in the grid bar nearest the axial centerline of the reactor is considered. The grid consists of two regions, the upper region being that where the graphite core blocks dowel-shaped ends terminate. The internal heat generation rate is a maximum in the upper region. It is assumed to be uniform throughout and all heat generated is transferred laterally.

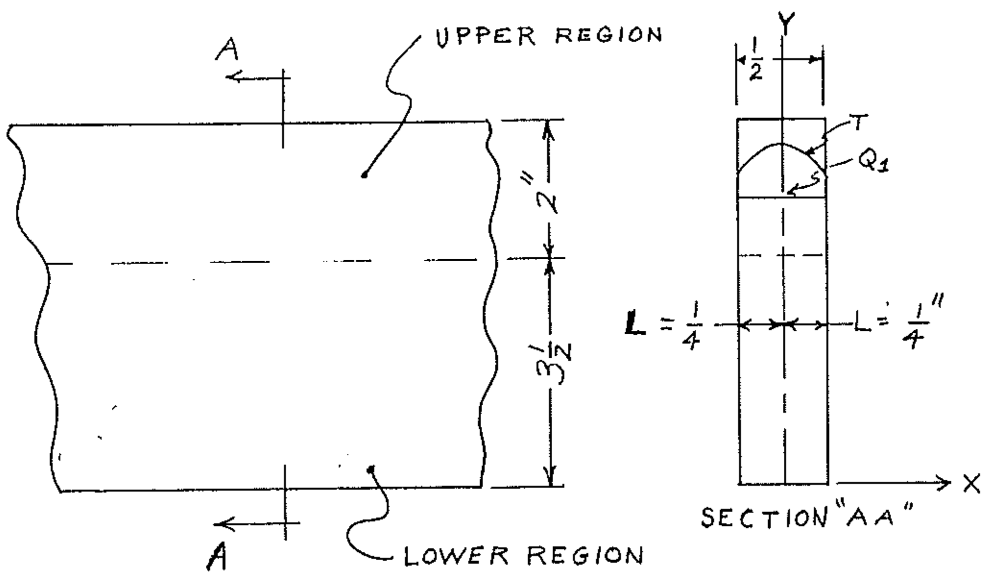


Fig. 12. Thermal Stress Model of Support-Grid Bar

A. The thermal stress is given by the following equation.[4]

$$\sigma = \frac{2}{3}\ \frac{\alpha E Q_1 L^2}{(1-\nu)\cdot 2\cdot K}$$

B. σ = Stress at x = L

$E = 24.7 \times 10^6$

$K = 12.68$ Btu-ft/hr-ft^2-°F

$Q_1 = 125.4$ Btu/in^3-hr (in upper region)

$L = .25$ in.

$\alpha = 8 \times 10^{-6}$ in./in.-°F

$\nu = .3$

C. $$\sigma = \frac{8 \times 24.7 \times 125.4 \times (.25)^2 \times 12}{.7 \times 12.68 \times 3} = 720 \text{ psi}$$

3. Core support-grid supporting ring fasteners

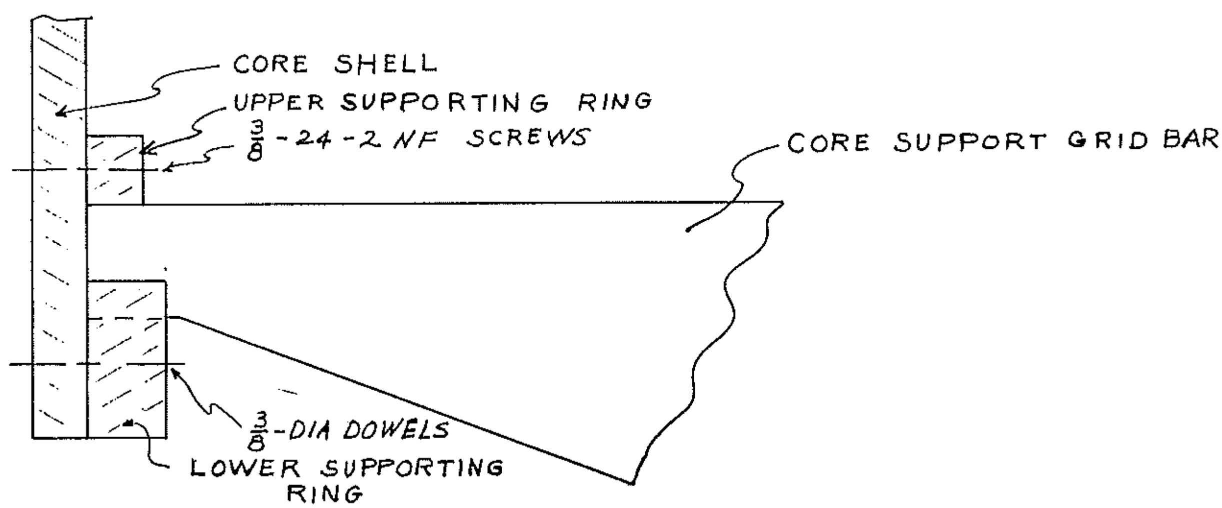


Fig. 13. Location and Arrangement of Support-Grid Support-Rings and Fasteners

A. Load on the upper ring

a. $F_B = V_1 (\rho_2 - \rho_1)$

b. F_B = Buoyancy force per core block

$V_1 = 211$ in.^{3} volume of one core block

$\rho_1 = 117.5$ lb/ft^3 density of core block

$\rho_2 = 154$ lb/ft^3 density of fuel salt

c. $F_B = \frac{211}{1728}(154 - 117.5) = 4.5$ lb per block

d. $F_P = (\Delta P_1 + \Delta P_2)\ \frac{.785\ D^2 \times \rho_2}{1728}$

e. F_P = Force due to pressure drop across graphite lattice and core blocks

ΔP_1 = .56 in. pressure drop across core blocks

ΔP_2 = 5.8 in. pressure drop across graphite lattice

D = 55.5 in. core shell I.D.

f. $F_P = (.56 + 5.8) \times .785 \times 55.5^2 \times \frac{154}{1728} = 1368$ lb

g. $F_T = F_P + 600\ F_b$

h. F_T - total load on upper support - ring fasteners

i. $F_T = 1368 + 600 \times 4.5 = 4068$ lb

B. Mechanical stress in upper support-ring fasteners

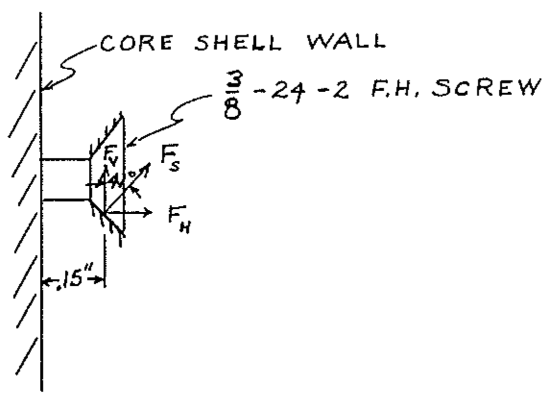


Fig. 14. Loading on the Upper Support-Ring Fasteners

The upper support ring has cross section dimensions of 3/8 in. x 5/8 in. and is fastened with 60 screws. The load per screw is determined as follows.

a. $F_V = \frac{F_T}{60}$

$F_V = \frac{4068}{60} = 67.8$ lb per screw

b. $F_H = F_V \tan 41°$

$F_H = 67.8 \times .869 = 60.9$ lb

Combined axial stress in the screw

c. $S_a = \frac{M}{I/c} + \frac{F_H}{A}$

$S_a = \frac{67.8 \times (.15/2)}{.098(.335)^3} + \frac{60.9}{.0878} = 2074$ psi

The shear stress is

d. $S_s = \frac{F_V}{A}$

$S_s = \frac{67.8}{.0878} = 772$ psi

The maximum normal stress is

e. $\sigma_n = \frac{S_a}{2} + \sqrt{\left(\frac{S_a}{2}\right)^2 + S_s^{\ 2}}$

$\sigma_n = \frac{2074}{2} + \sqrt{\left(\frac{2074}{2}\right)^2 + 772^2}$

$\sigma_n = 1037 + 1294 = 2331$ psi

The maximum shear is

f. $\sigma_s = \sqrt{\left(\frac{S_a}{2}\right)^2 + S_s^{\ 2}}$

$\sigma_s = \sqrt{\left(\frac{2074}{2}\right)^2 + 772^2} = 1294$ psi

C. Mechanical stress in lower supporting ring fastener

The lower supporting ring is fastened by sixty-eight 3/8-in.-dia dowels, plug welded on each end.

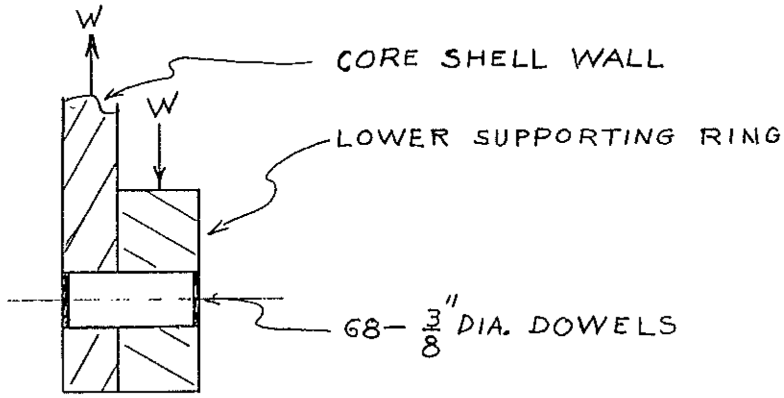


Fig. 15. Loading on the Lower Supporting Ring Fastener

The shear stress is

$$S = \frac{W}{A}$$

$$S = \frac{10{,}000}{68 \times .785^2 \times .375^2} = 1332 \text{ psi}$$

4. Core shell support-flange attached to the shell

The flange is attached to the core shell by seventy-two 3/8-in.-dia dowels, plug welded at each end.

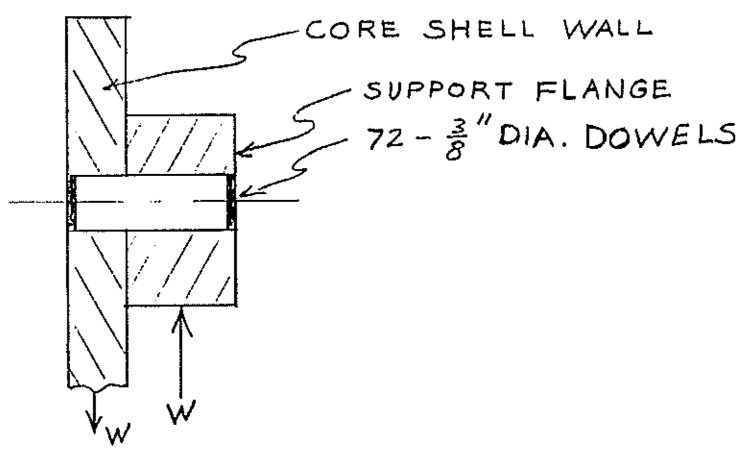


Fig. 16. Loading on the Shell Support - Flange Attached to the Shell.

The shear stress is

$$S = \frac{W}{A}$$

$$S = \frac{11,600}{72 \times .375^2 \times .785} = 1460 \text{ psi}$$

5. Core shell support-flange attached to the reactor vessel shell

The flange is attached to the vessel by two circumferential fillet welds.

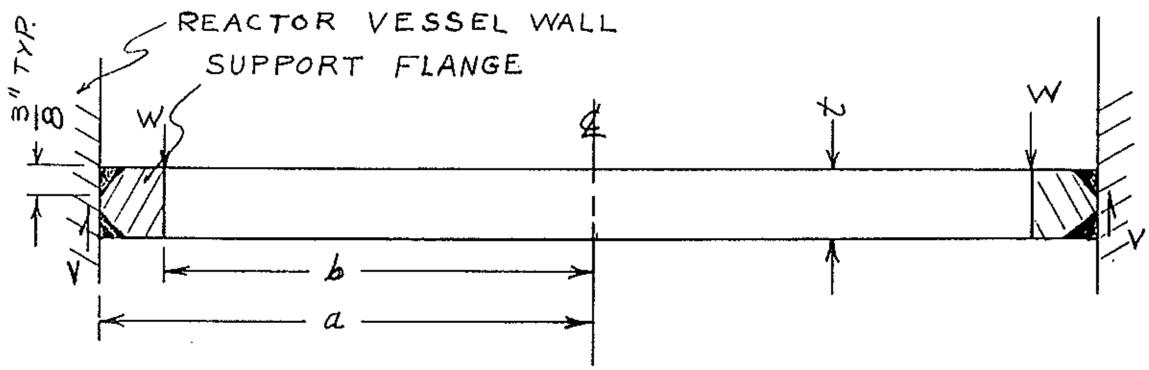


Fig. 17. Loading on the Shell Support Flange Attached to the Vessel Wall.

A. The radial stress is given by the following formula.[8]

$$S_r = \frac{3W}{2\pi t^2}\left[1 - \left(\frac{2mb^2 - 2b^2\,(m+1)\,\ln\frac{a}{b}}{a^2(m-1) + b^2(m+1)}\right)\right]$$ Roark page 199

B. W = 11,600 lb, evenly distributed load

t = 1 in.

a = 29 in.

b = b = 28.125

ν = .3

m = $1/\nu$ = 3.333

C. $S_r = \frac{3 \times 11,600}{2\pi \times 1.0002}\left[1 - \left(\frac{6.666 \times 28.125^2 - 2 \times 29^2(4.333)\ \ln \frac{29}{28.125}}{29^2(2.333) + (29.125)^2(4.333)}\right)\right]$

$S_r = 5536\ (1 - 0.937) = 349$ psi

D. The tangential is given by the following.[8]

$$S_t = \frac{3W}{2\pi mt^2}\left[1 + \left(\frac{ma^2(m-1) - mb^2(m+1) - 2(m^2-1)a^2 \ln \frac{a}{b}}{a^2(m-1) + b^2(m+1)}\right)\right]$$

Substituting from the values above

E. $S_t = 1661 \left[1 - 1.0124\right] = -20$ psi

F. The shear stress at the outer periphery is determined by the following.

$$S_s = \frac{W}{\pi a \times 2 \times 3/8}$$

$$S_s = \frac{11,600}{\pi \times 29 \times .75} = 117 \text{ psi}$$

The internal heat generation in the core shell support flanges being appreciably smaller than the values reported for the reactor vessel shell and reactor core shell results in insignificant thermal stresses.

6. Thermal stress in the core shell

The region at the midplane where the internal heat generation rate is maximum is considered. The heat generation rate is assumed to be uniform and the heat is transferred radially outward.

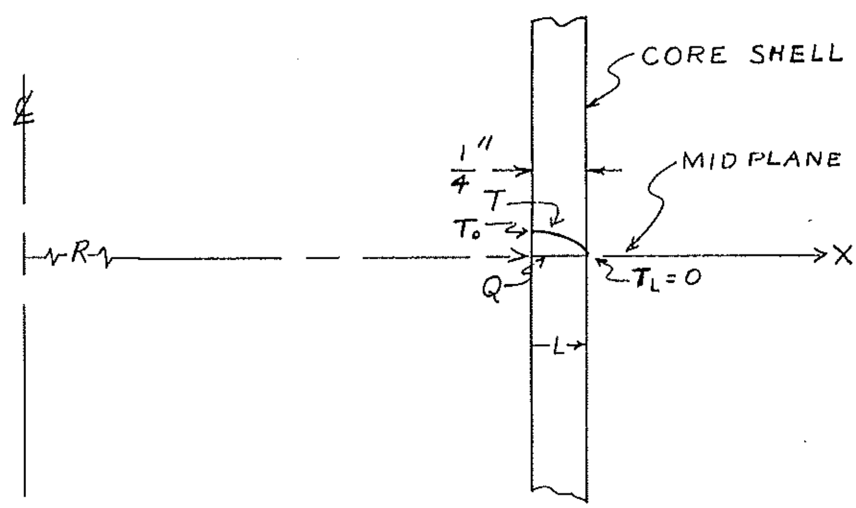


Fig. 18. Core Shell Thermal Stress Model

A. $\frac{\partial^2 T}{\partial x^2} = -\frac{Q}{K}$ (ref. 4)

B. $T = \Delta T = \frac{QL^2}{2K}$

C. $\sigma = \frac{2}{3}\,\frac{\alpha EQL^2}{(1-\nu)2K}$ (ref. 4)

Substituting from the values, section IV-2B, except Q = 7.34 Btu/hr-in.^{3}:

D. $\sigma = \frac{8 \times 24.7 \times 7.34 \times (.25)^2 \times 12}{.7 \times 3 \times 12.68} = 42$ psi

V. Design of the Graphite Lattice

The graphite lattice rests on the grid support bars resulting in the longest unsupported length being in the lower lattice bars between adjacent grid support bars. Figure 19 depicts a typical segment in the lower lattice bars with the loads from the core block (2 blocks loading each segment) considered as interrupted, distributed loads.

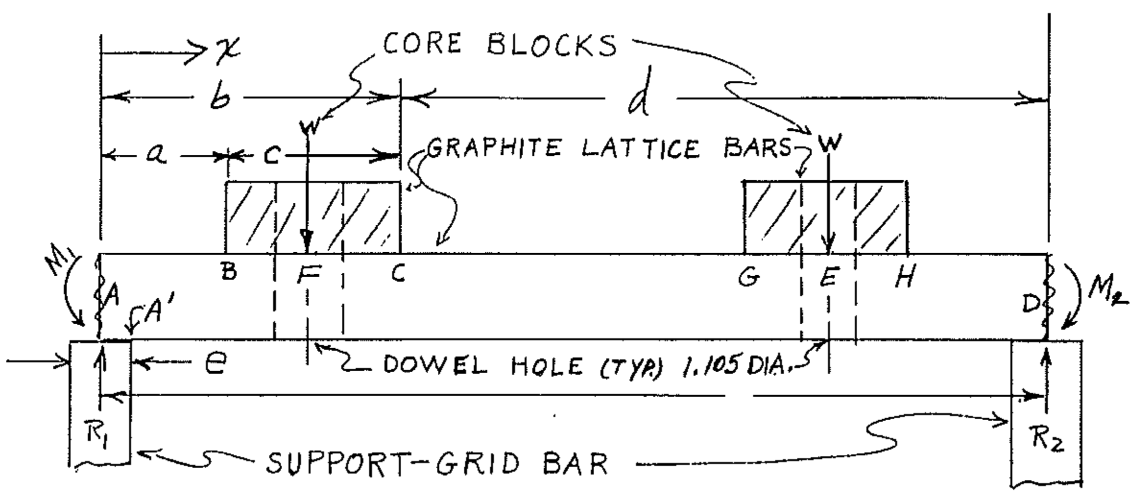


Fig. 19. Typical Segment of Lower Lattice Bars

1. A. $a = .187$ in.
   $b = 1.812$ in.
   $c = 1.625$ in.
   $d = 3.187$ in.
   $\ell = 4.000$ in.
   $W = 14$ lb, weight of one core block

B. The moment at A due to the load on the interval B-C is given by the following equation.[8]

$$M_1 = -\frac{W}{24\ell}\left(\frac{24d^3}{\ell} - \frac{6bc^2}{\ell} + \frac{3c^3}{\ell} + 4c^2 - 24d^2\right)$$

$$M_1 = -\frac{14}{24 \times 4}\left(\frac{24 \times 3.187^3}{4} - \frac{6 \times 1.812 \times 1.625^2}{4} + \frac{3 \times 1.625^3}{4} + 4 \times 1.625^2 - 24 \times 3.187^2\right)$$

$$= -.1458\ (-42.941) = 6.26 \text{ in.-lb}$$

C. The moment at D due to the load on the interval B-C is the same as the moment at A due to the load on the interval G-H and is given by the following equation.[8]

$$M_2 = \frac{1}{24}\frac{W}{\ell}\left(\frac{24d^3}{\ell} - \frac{6bc^2}{\ell} + \frac{3c^3}{\ell} + 2c^2 - 48d^2 + 24\ d\ell\right)$$

$$M_2 = .1458\ (194.22 - 7.178 + 3.219 + 5.282 - 487.536 + 305.952)$$

$$M_2 = .1458 \times 13.959 = 2.035 \text{ in.-lb}$$

D. $M = M_1 + M_2 = 6.26 + 2.04 = 8.3$ in.-lb

E. The moment at F where $x = 1$ due to the load on the interval B-C is:[8]

$$M_3 = -M + R_1x - 1/2\ \frac{W(x - a)^2}{C}$$

$$= -8.3 + 14 - \frac{.5 \times 14(.813)^2}{1.625} = 2.85 \text{ in.-lb}$$

F. The moment at E due to the load on the interval B-C is the same as the load at F due to the load on the interval G-H and is given by the following with

$x = 3$ in.

$M_4 = - M + R_1 x - W(x - \ell + d)$ (ref. 8)

$M_4 = - 8.3 + 14 \times 3 - 14(3 - 4 + 3.187) = 3.082$ in.-lb

2. The stresses at critical points are as follows.

A. The bending stress at F is:

$$S_B = \frac{Mc}{I}$$

$$S_B = \frac{(M_3 + M_4)(.925/2)}{\frac{1}{12}(1.625 - 1.105)(.925)^3} = 80 \text{ psi}$$

B. The bending stress at A is

$$S_B = \frac{Mc}{I}$$

$$S_B = \frac{8.3(.925/2)}{\frac{1}{12}(1.625)(.925)^3} = 36 \text{ psi}$$

The shearing force at A' is $R_1 = 14$ lb and at F is $R_1/2 = 7$ lb. Thus the highest shear stress is at F due to the greatly reduced area.

C. The shear stress at F is (see Figs. 19 and 20).

$$S_s = \frac{R_1/2}{A}$$

$$S_s = \frac{14/2}{(1.625 - 1.105)\ .925} = 14.5 \text{ psi}$$

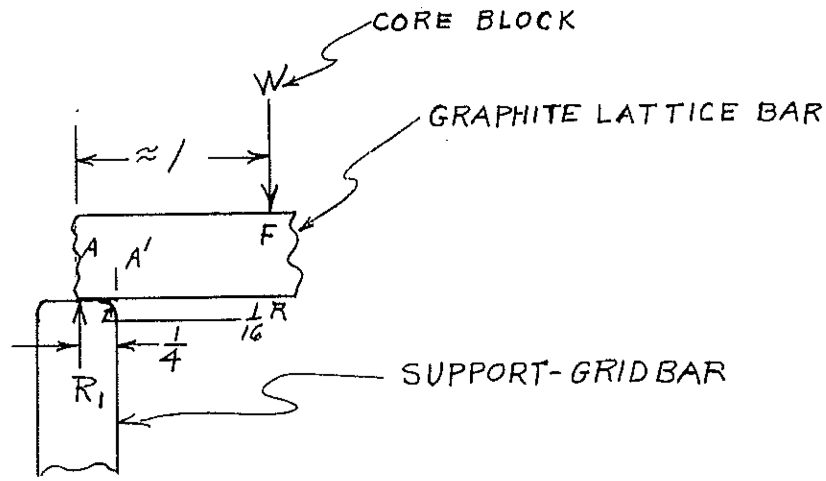


Fig. 20. Graphite Lattice Bar Shear and Compressive Stress Model (from Fig. 19)

D. The compressive stress at A is

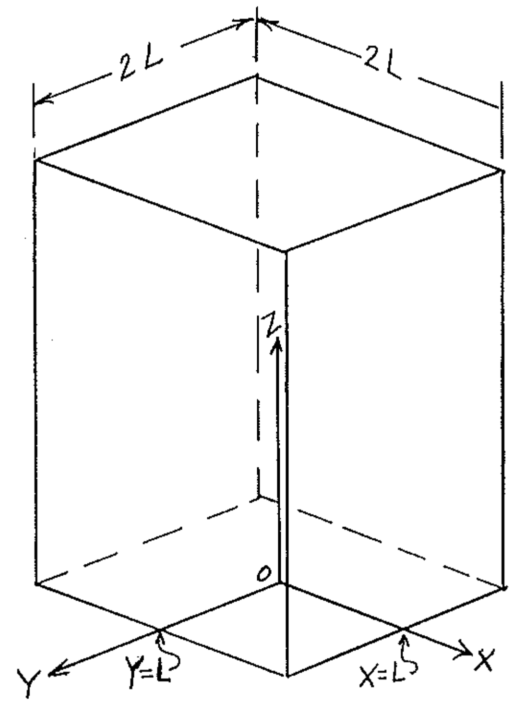
$$S_c = \frac{R_1}{A}$$

$$S_c = \frac{14}{(.25 - .062)\ 1.625} = 47 \text{ psi}$$

VI. Critical Thermal and Mechanical Stress in the Core Blocks

B. W. Kinyon[9] reported the effects of irradiation on the core blocks and the stress that would result from restraining the bowing.

1. Thermal stress due to internal heat generation is determined as follows.



$$x = y = L$$

$$T = 0$$

$$x = y = 0$$

$$\frac{\partial T}{\partial x} = \frac{\partial T}{\partial y} = 0$$

$$\frac{\partial^2 T}{\partial x^2} = \frac{\partial^2 T}{\partial y^2}$$

Fig. 21. Graphite Core Block Thermal Stress Model

Assume uniform internal heat generation and all surfaces at the same temperature.

A. $\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = \frac{2\,\partial^2 T}{\partial x^2} = -\frac{Q}{K}$ (ref. 4)

B. $T = -\frac{Qx^2}{4K} + \frac{QL^2}{4K} = +\frac{Q}{4K}(-x^2 + L^2)$

C. $\sigma = \frac{\alpha EQ}{4K(1-\nu)}\left[x^2 - \frac{L^2}{3}\right]$ (ref. 4)

D. $L = \frac{.8}{12}$ ft

$K = 12$ Btu-ft/hr-ft^2-°F

$E = 3 \times 10^6$ psi

$Q = .05 \times 1.033 \times 10^6$ Btu/hr-ft^3

$\nu = .3$

$\alpha = .56 \times 10^{-6}$ in./in.-°F

E. At $x = L$, the thermal stress is

$$\sigma = \frac{\alpha EQ}{4K(1-\nu)}\left[\frac{2}{3}L^2\right]$$

$$\sigma = \frac{.56 \times 3 \times .05 \times 1.033 \times 10^6}{4 \times 12 \times .7}\left[\frac{2}{3}\left(\frac{.8}{12}\right)^2\right]$$

$$\sigma = 7.6 \text{ psi}$$

2. The mechanical stress in the core block anchoring extension is determined as follows.

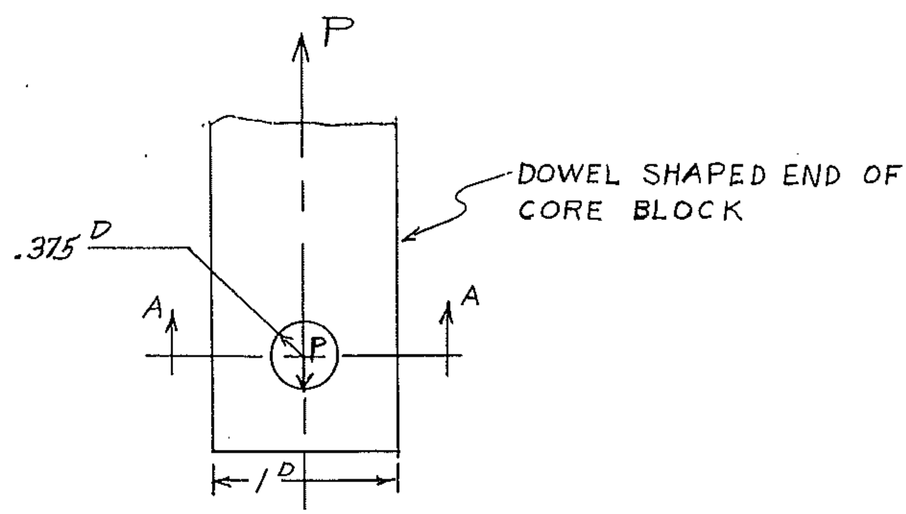


Fig. 22. Loading on the Dowel-Shaped End of a Core Block

A. $S_T = \frac{PK}{A}$

B. S_T = Tensile stress at A-A

P = Load due to flotation and pressure drop across core

A = Area at A-A

K = Stress concentration factor (2.1)

C. $P = F_B + F_P$ per block (see section III-3)

$P = 4.5 + 6.36\ (2 \times 2) \times \frac{154}{1728} = 6.76$ lb per block

Use 7 lb

D. $S_T = \frac{7 \times 2.1}{.785 - .375} = 37$ psi

VII. Mechanical Design of the Support Lugs

The reactor is suspended from the top section of the thermal shield by twelve 1-1/4-in.-dia rods. An eye on the lower end of the rod is engaged by a pin through lugs welded to the reactor vessel shell (Fig. 23).

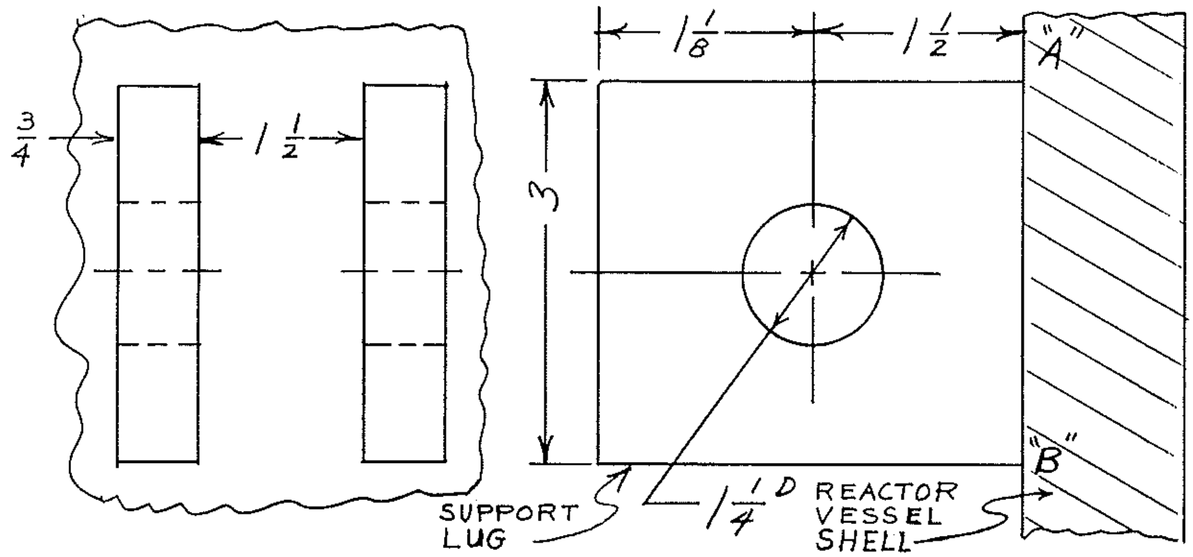


Fig. 23 Sketch of the reactor vessel support lugs

The principal stresses at points A and B are determined as follows:

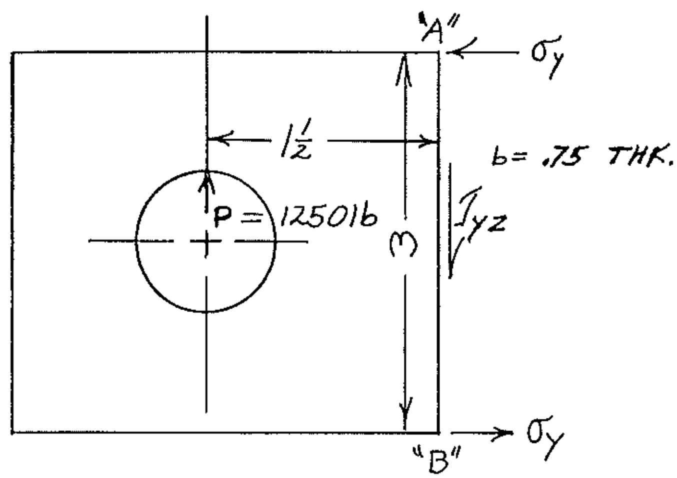


Fig. 24 Loading on the reactor support lugs

1. Bending and shear stresses where the lugs join the vessel,

A.
$$\sigma_y = \frac{6M}{bh^2}$$

$$\sigma_y = \frac{6(1250 \times 1.5)}{.75 \times 3^2} = 1667 \text{ psi}$$

B.
$$\tau_{yz} = \frac{P}{bh}$$

$$\tau_{yz} = \frac{1250}{.75 \times 3} = 556 \text{ psi}$$

2. In the vessel wall the stresses due to design pressure are,

A. circumferential

$$\sigma_x = \frac{PD}{2t}$$

$$\sigma_x = \frac{50 \times 59}{2} = 1475 \text{ psi}$$

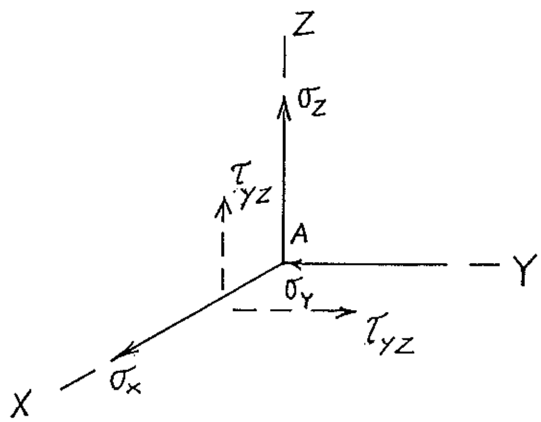
B. longitudinal

$$\sigma_z = \frac{1}{2}\,\sigma_x$$

$$\sigma_z = \frac{1475}{2} = 738 \text{ psi}$$

3. The principal stresses at A and B are given by the following equation,[5]

$$S^3 - (\sigma_x + \sigma_y + \sigma_z)S^2 + (\sigma_x\sigma_y + \sigma_y\sigma_z + \sigma_x\sigma_z - \tau_{yz}{}^2)S$$
$$-(\sigma_x\sigma_y\sigma_z - \sigma_x\tau_{yz}{}^2) = 0$$



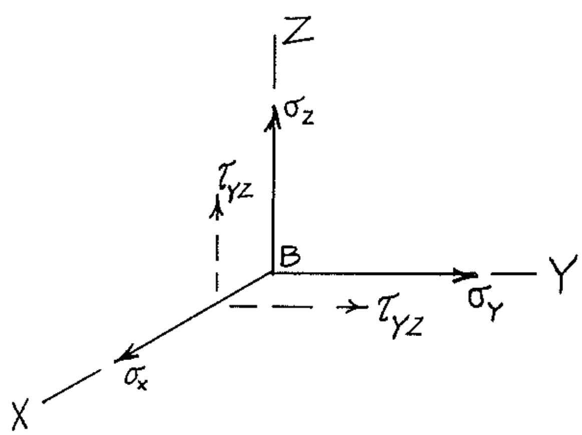
σ_x = 1475 psi
σ_y = -1667 psi
σ_z = 738 psi
τ_{yz} = 556 psi

Fig. 25 Stresses in the vessel wall at A

4. The principal stresses at A are,

$$S^3 - 546S^2 - 2,600,521\ S + 2,270,588,450 = 0$$

$S \approx -1790$ psi
≈ 1472 psi
≈ 560 psi



σ_x = 1475 psi
σ_y = 1667 psi
σ_z = 738 psi
τ_{yz} = 556 psi

Fig. 26 Stresses in the vessel wall at B

5. The principal stresses at B are,

$$S^3 - 3880\ S^2 + 4{,}468{,}485\ S - 1{,}358{,}637{,}250$$

$$\begin{aligned} S &\approx 1920 \text{ psi} \\ &\approx 1485 \text{ psi} \\ &\approx 475 \text{ psi} \end{aligned}$$

6. The contact stresses where the support rod engages the support lugs is given by the following equation,[8]

$$S_c = 0.591 \sqrt{PE\ \frac{D_1 - D_2}{D_1 \times D_2}}$$

D_1 = Hole dia P = Load/linear in.
D_2 = Pin dia $E = 24.7 \times 10^6$

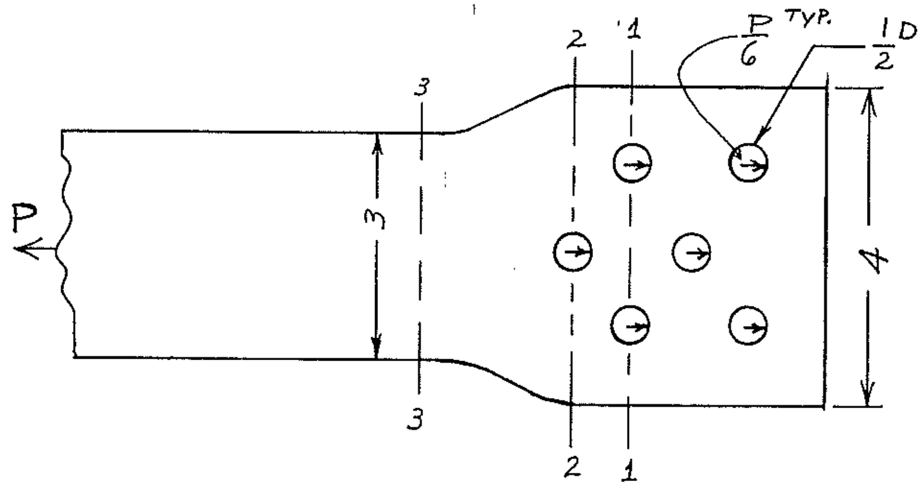
$$S_c = 0.591 \sqrt{\frac{1250}{.75} \times 24.7 \times 10^6\ \frac{(1.265 - 1.25)}{1.265 \times 1.25}}$$

$$S_c = 0.591 \sqrt{3.90 \times 10^8} = 11{,}676 \text{ psi}$$

This contact stress is less than two-thirds the yield stress but considerably higher than two-thirds the stress necessary to produce 0.1% creep in 10,000 hr. As a result a minute amount of creep will occur, which increases the contact area, rapidly lowering the stress to an acceptable value.

VIII. Molybdenum Band Design

The core blocks are restrained from excessive bowing by five molybdenum bands spaced over an 18-in.-long section symmetrical about the midplane. The bands are joined by rivets with a single cover plate.



Band 1/8-in. thick

Fig. 27 Loading in molybdenum bands

Assume uniform loading on the rivets. The various stresses are determined as follows:

1. $$P = \frac{11000}{5} = 2200 \text{ lb} \qquad \text{(ref. 9)}$$

2. A. Bearing stress between rivets and bands:

$$S_B = \frac{P/6}{A_1}$$

$$S_B = \frac{2200}{6 \times 1/2 \times 1/8} = 5870 \text{ psi}$$

B. Shearing stress in the rivets:

$$S_s = \frac{P/6}{A_2}$$

$$S_{s_1} = \frac{2200}{6 \times .785 \times .5^2} = 1870 \text{ psi}$$

C. Shearing stress in the bands:

$$S_{s_2} = \frac{P/6}{A_3}$$

$$S_{s_2} = \frac{2200}{6 \times 1/8 \times 2 \times 1.625} = 903 \text{ psi}$$

D. Tensile stress at 1-1, 2-2 and 3-3 (Fig. 27):

a. $$S_{t_1} = \frac{(P - P/6)}{A_4}$$

$$S_{t_1} = \frac{2200 - 2200/6}{1/8(4 - 1/2 - 1/2)} = 4888 \text{ psi}$$

b. $$S_{t_2} = \frac{P}{A_5}$$

$$S_{t_2} = \frac{2200}{1/8(4 - 1/2)} = 5029 \text{ psi}$$

c. $$S_{t_3} = \frac{P}{A_6}$$

$$S_{t_3} = \frac{2200}{1/8 \times 3} = 5867 \text{ psi}$$

IX. Cooling of the Control Rods

In MSR-61-54 J. H. Westsik reported the highest temperature and the temperature distribution that would occur in the rod with no cooling air. The maximum temperature was reported to be 1427°F at the inside can of the B_4C rod segment. Assuming the flexible hose on which the control rod segments are threaded is at a uniform temperature of 1275°F, while air is flowing, the temperature rise of the air and the amount of air required are determined. The rod is considered to consist of six sections and mean values of the air properties (c_p, μ, k, etc.) for each section are used.

1. The following assumptions and conditions are used.

   A - inside area of a section of the flexible hose

   c_p - mean specific heat

   G - mass velocity

   D = .625 in. ID of the flexible hose

   h - film coefficient, metal-to-air in the flexible hose

   L = 66 in., length of the rod

   q = 2000 Btu/hr heat to be removed per rod

   Δt - temperature difference

   w - amount of air required - lb/hr

   Δt_m - log mean Δt of the air through one section of the rod

2. Assuming an air outlet temperature at the lower end of the rod of 1000°F, the amount of air required is:

$$w = \frac{q}{c_p \Delta t}$$

$$w = \frac{2000}{.25(1000 - 200)} = 10 \frac{lb}{hr} \quad \text{per rod}$$

3. Considering the rod as consisting of six sections, the temperature rise of the air is determined from the following equations:

   A. $h A \Delta t_m = w c_p(t_{b'} - t_{a'})$ See Fig. 28

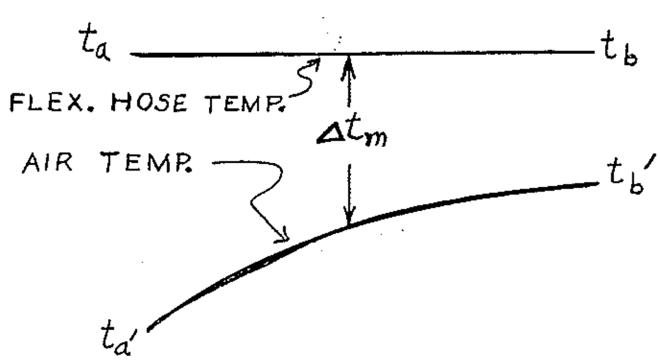


Fig. 28 Temperature distribution in a section of the control rod

B. $$\Delta t_m = \frac{(t_a - t_{a'}) - (t_b - t_{b'})}{\ln \dfrac{(t_a - t_{a'})}{(t_b - t_{b'})}}$$ (Ref. 10)

C. $$h = \frac{.023 \left[1 + (D/L)^{.7}\right] G^{.8} c_p}{(Pr)_b^{.667} \left(\dfrac{\mu_w}{\mu_b}\right)^{0.14} \left(\dfrac{D}{\mu_b}\right)^{.2}}$$ (Ref. 10)

D. Substituting from above and simplifying results in the following:

a. $$h = \frac{16.15\ cp}{Pr^{.667} \left(\dfrac{\mu_w}{\mu_b}\right) \left(\dfrac{.052}{\mu_b}\right)^{.2}}$$

b. $$(t_b - t_{b'}) = \ln^{-1}\left[\ln(t_a - t_{a'}) - \frac{h}{c_p}\ .01637\right]$$

From the above equations the temperature of the air at the lower end of the rod when fully inserted is estimated to be $1180^{\circ}F$, the heat removed 2350 Btu/hr with an air flow rate of 10 lb/hr.

Should the air supply fail, all the heat generated in the rod and thimble must be transferred to the fuel salt by conduction from the thimble. At the midplane, where the heat generation rate is $2.5\ w/c^3$, the maximum temperature difference (Δt) between the mean fuel temperature and thimble is determined in the following analysis by W. H. Ford.

The thimble surface is considered as a flat surface with a mirror image equidistant from an adiabatic surface (see Fig. 29).

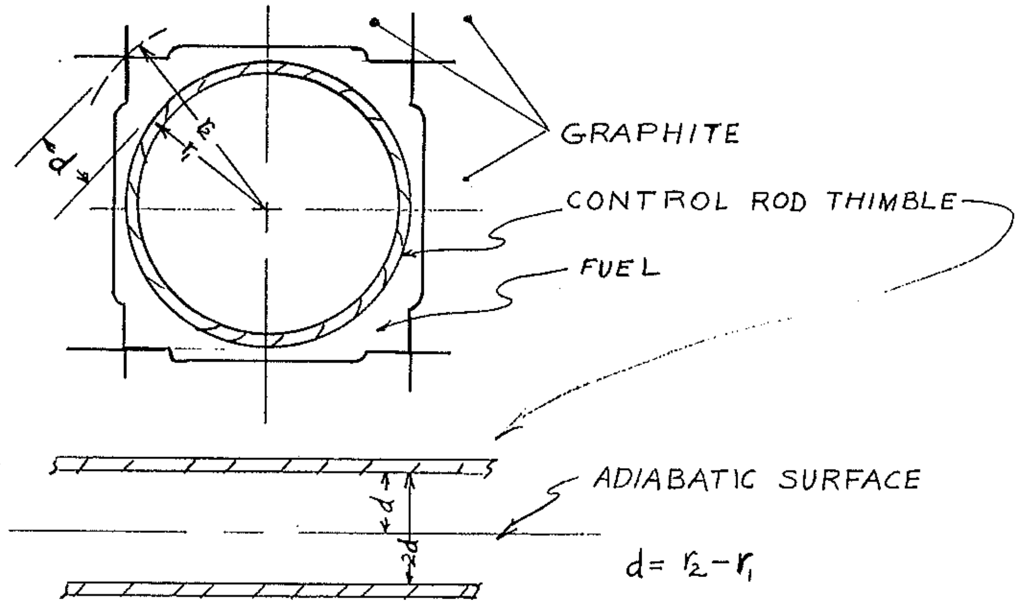


Fig. 29 Cross Section of Control Rod Thimble in Graphite Matrix and Mathematical Model

4. The temperature difference is determined as follows:[11]

$$\frac{\Delta t}{\frac{Q_f \, d^2}{k}} = \frac{17F^1 - 14}{35}$$

where $F^1 = 1 - \frac{1}{Q_f d} \frac{dq}{dA}$

$d = 0.414$ in. Distance from thimble surface to adiabatic surface

$\frac{dq}{dA} = 12.27 \frac{\text{Btu-in.}}{\text{hr-in.}^2}$ Heat flux per inch of rod length at the midplane

$k = 3.5$ Btu-ft/hr-ft^2 °F Thermal conductivity of the fuel

$Q_f = 2985$ Btu/hr-in.^{3} Heat generation rate in the fuel around the thimble

Δt = Temperature difference between the thimble surface and mean fuel temperature

$\Delta t \approx 12^{\circ}$F

Appendix A

DESIGN DRAWINGS OF THE MSRE CORE NO. 1

Drawing Number	Title
E-BB-B40400	Reactor Assembly Plan
E-BB-B40401	Reactor Assembly Section, Sheet No. 1
E-BB-B40402	Reactor Vessel Shell, Flow Distributor and Bottom Head Assembly
D-BB-B40403	Reactor Vessel Bottom Head Assembly
D-BB-B40404	Reactor Vessel Head Top
D-BB-B40405	Reactor Vessel Head - Bottom Details
D-BB-B40406	Reactor Vessel Shell
D-BB-B40407	Reactor Vessel Flow Distributor Assembly
D-BB-B40408	Reactor Vessel Flow Distributor Details, Sheet No. 1
D-BB-B40409	Reactor Vessel Flow Distributor Details, Sheet No. 2
D-BB-B40410	Reactor Core Shell Assembly
D-BB-B40411	Reactor Core Shell Details, Sheet No. 1
D-BB-B40412	Reactor Core Shell Details, Sheet No. 2
D-BB-B40413	Reactor Core Grid Assembly
D-BB-B40414	Reactor Core Grid Details, Sheet No. 1
D-BB-B40415	Reactor Core Grid Details, Sheet No. 2
D-BB-B40416	Reactor Core Block Assembly Plan
D-BB-B40417	Reactor Core Block Details, Sheet No. 1
D-BB-B40418	Reactor Core Block Details, Sheet No. 2
D-BB-B40419	Reactor Core Block Details, Sheet No. 3
D-BB-B40420	Reactor Core Lattice Block Assembly
D-BB-B40421	Reactor Core Lattice Block Details, Sheet No. 1
D-BB-B40422	Reactor Core Lattice Block Details, Sheet No. 2
D-BB-B40424	Reactor Core Block Centering Bridge
D-BB-B40425	Moly-Band Assembly and Details
D-BB-B40427	Reactor Core Flow Restrictor Ring - Assembly and Details
D-BB-B40429	Reactor Core Shell Details, Sheet No. 3
D-BB-B40491	Reactor Core Block Details, Sheet No. 4
D-BB-B40492	Upper Moly-Band Assembly and Details
D-BB-B40493	Reactor Core Block and Grid Assembly - Section
D-BB-B40494	Reactor Core Block and Core Shell Assembly
D-BB-B40581	Reactor Core Block Details, Sheet No. 5
D-BB-B40585	Reactor Access Plug Air Cooling Manifold Assembly
D-BB-B40586	Reactor Access Plug Air Cooling Manifold Details
D-BB-B40587	Reactor Graphite Sampler Assembly
D-BB-B40588	Reactor Graphite Sample Access Plug, Air Inlet and Outlet Assys. and Details
D-BB-B40589	Reactor Graphite Sample Retaining Rod, Support Rack and Sample Assys. and Details
D-BB-B40590	Reactor Control Rod Guide Bar Assembly and Details
E-BB-B40594	Reactor Assembly Section, Sheet No. 2
E-BB-B40595	Reactor Access Nozzle and Access Plug Assemblies

D-BB-B40596	Reactor Nozzle and Access Plug Cooling Details
E-BB-B40597	Reactor Nozzle Air Cooling Manifold Assembly and Details
E-BB-B40598	Reactor Control Rod Thimble Assembly and Details
E-BB-B40599	Reactor Access Plug Shell Assembly and Details
D-BB-B40600	Reactor Control Rod Assemblies and Details

References

1. E. S. Bettis, Control Rod Constants, MSR-61-53 (May 17, 1961).

2. R. J. Kedl, Predicted Flow Distribution in the MSRE Core (A section of this report).

3. ASME Boiler and Pressure Vessel Code, Section VIII, Unfired Pressure Vessels (1959).

4. S. Glasstone et al, Principles of Nuclear Reactor Engineering, D. Van Nostrand Co., New York (1955).

5. S. Timoslenko and J. N. Goodier, Theory of Elasticity, 2nd ed., McGraw-Hill Book Co., New York (1951).

6. C. W. Nestor, Personel Communication (June 1961).

7. J. C. Moyers, Personel Communication (April 1961).

8. R. J. Roark, Formulas for Stress and Strain, 3rd ed., McGraw-Hill Book Co., New York (1954).

9. B. W. Kinyon, Effects of Graphite Shrinkage in MSRE Core, ORNL CF 60-9-10 (September 2, 1960).

10. W. H. McAdams, Heat Transmission, 3rd ed., McGraw-Hill Book Co., New York (1954).

11. L. D. Palmer and H. F. Poppendiek, Forced Convection Heat Transfer Between Parallel Plates in Annuli with Volume Heat Sources within the Fluid, ORNL 1701 (May 11, 1954).

II.